UNIVERSITY OF ILLINOIS ENGINEERING EXPERIMENT STATION

Bulletin Series No. 401

COMPARATIVE PERFORMANCES OF A WARM-AIR CEILING PANEL SYSTEM AND A CONVECTION SYSTEM

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ABSTRACT

This bulletin presents the first results obtained in Warm-Air Heating Research Residence No. 2, which was completely equipped specifically for research in warm-air heating by the National Warm Air Heating and Air Conditioning Association. This Residence, which was completed in June 1947, was a one-story structure of frame construction with a large amount of glass exposure and with a full basement.

The convection system referred to in the bulletin was a conventional forced warm-air heating system which included an extended-plenum trunk duct and high-sidewall registers. The results obtained with this convection system were compared with those of a warm-air ceiling panel system in which no heated air was introduced into the rooms. The panel system utilized the 8-in.-deep air space provided by the open-web steel joists used in the ceiling. The lower panel surface, or ceiling of the first-story rooms, was constructed of pressed cement-asbestos millboard panels. The upper surface of the panel space consisted of ½-in. gypsum sheets, above which was placed a 3⅝-in. thickness of batt-type insulation in order to minimize the heat loss from the panel space to the attic. Relatively low air-flow rates and low air velocities were utilized in the panel space.

The two systems were connected to the same furnace and were operated alternately throughout the heating season, so that the performance of each system was obtained over a wide range of weather conditions. The performance of the panel system is representative only of the warm-air ceiling panel as installed and does not necessarily represent that which might be obtained with other heat transfer mediums or other locations of the panel. As far as air temperatures and average surface temperatures of the rooms were concerned, the performances of the panel system and the convection system were remarkably alike. Both the room-air temperature differentials in the living zone and the mean radiant temperatures were only slightly in favor of the panel system. The mean radiant temperature observed at the center of the living room by means of a thermo-integrator was only 0.5 F higher than the ambient air temperature for the panel system. Corresponding observations made with the convection system indicated that the mean radiant temperature was only 0.3 F lower than the ambient air temperature.

The similarity of performances of the panel and convection systems may be attributed largely to the effect of direct and indirect heat regains which occurred within the entire structure. These regains also accounted for deviations which occurred between design values and actual performance values.

Excellent automatic control of room-air temperatures was obtained for both the convection and the panel systems with a conventional, heat-anticipating room thermostat for which a minimum differential setting was used. In the case of the convection system the use of the minimum differential setting of the room thermostat was found to be more effective for obtaining close control of room-air temperatures than was the setting of the fan switch.

Under normal operation with the panel system no temperature over-runs or thermal lags were experienced. However, under conditions of night set-back of the thermostat a much slower recovery rate during the morning pickup period was obtained for the panel system than for the convection system.

The fuel consumption for the panel system was higher than that for the convection system. It was evident from this difference in fuel consumption that adequate insulation of ducts in an attic as well as the top side and exposed edges of a panel space is essential. The results obtained with continuous operation of the blower showed a lower fuel but a higher electrical consumption than those for intermittent operation.

Although satisfactory temperature conditions were obtained in the first-story rooms when no heat was introduced into the basement, marked improvements were noted when the basement was heated, as a result of panel-heating effects from the floor. These improved temperature conditions were gained at the expense of a 10 percent increase in fuel consumption.

It was concluded that for the purpose of selecting the maximum fuel-input rate the total heat loss from the structure, including the basement, should have been considered regardless of whether or not the basement was heated.

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I. INTRODUCTION

1. Preliminary Statement

This bulletin is a report of the results obtained during the 1947-48 and 1948-49 heating seasons in Warm-Air Heating Research Residence No. 2. This Residence, which was completed in June 1947, was built, furnished, and completely equipped specifically for research in warm-air heating by the National Warm Air Heating and Air Conditioning Association. It replaced the original Research Residence[(1)]* in which investigations of warm-air heating systems and summer-cooling systems were conducted from 1924 to 1946.

This investigation was conducted under the terms of a cooperative agreement made in 1918 between the National Warm Air Heating and Air Conditioning Association and the Engineering Experiment Station of the University of Illinois. In this cooperative research agreement the Association is represented by its Research Advisory Committee. During the period of investigation reported the committee consisted of sixteen men:

F. L. Meyer, Chairman, The Meyer Furnace Co., Peoria, Ill.

R. K. Becker, Ohio Valley Hardware and Roofing Co., Evansville, Ind.

J. B. Burrowes, Lau Blower Co., Dayton, Ohio.

K. T. Davis, Bryant Heater Division of Affiliated Gas Equipment, Inc., Cleveland, Ohio.

G. W. Denges, Williamson Heater Co., Cincinnati, Ohio.

R. S. Dill, National Bureau of Standards, Department of Commerce, Washington, D. C.

E. R. Downe, C. A. Olsen Manufacturing Co., Elyria, Ohio.

R. A. Gulick, May-Fiebeger Co., Newark, Ohio.

W. W. Johns, Johns and Son Furnace Co., Urbana, Ill.

Gordon Kinsman (deceased), Lau Blower Co., Dayton, Ohio.

C. W. Nessell, Minneapolis-Honeywell Regulator Co., Chicago, Ill.

F. J. Nunlist, L. J. Mueller Furnace Co., Milwaukee, Wis.

N. A. Palmer, Eureka-Williams Corp., Bloomington, Ill.

H. F. Randolph, International Heater Co., Utica, N.Y.

* Parenthesized superscript numbers refer to the corresponding entries in References.

F. W. Taylor, Canadian Chapter, National Warm Air Heating and Air Conditioning Association, Toronto, Ont.

H. Weyenberg, Holland Furnace Co., Holland, Mich.

2. Acknowledgments

The investigation was sponsored in part by the American Gas Association through its Committee on Domestic Gas Research as a PAR Plan activity of the Association (A.G.A. Project DGR-7-CH).

Acknowledgment is made to the manufacturers who cooperated by furnishing equipment used in the investigation.

3. Objectives of Investigation

The over-all objective of the investigations conducted in Research Residence No. 2 is to make thorough studies of the performance characteristics of warm-air heating systems, with emphasis on the evaluation of the comfort produced by these systems and of the cost of producing that comfort.

Specifically, the objectives of this particular investigation were as follows:

(a) To study the performance of a conventional forced warm-air or winter air conditioning system, referred to in this bulletin as a *convection system,* over a wide range of outdoor weather conditions. These results serve as a basis for comparison with any other system which might be investigated in the Residence.

(b) To study the performance of a warm-air ceiling panel system, referred to in this bulletin as a *panel system.*

(c) To compare the two systems as to general performance, relative comfort produced, and the cost of producing that comfort.

(d) To study each system with both continuous and intermittent blower operation.

(e) To study both systems with heat introduced into the basement.

(f) To study both systems with no heat introduced into the basement.

The study of the two systems with heat introduced into the basement was conducted during the 1947-48 heating season. The results are presented in the first part of this bulletin. As these results were analyzed it became apparent that a study of the systems with no heat introduced into the basement was necessary in order to complete the investigation. Therefore, such a study was conducted during the 1948-49 heating season, and the results are presented in the second part of the bulletin.

4. Glossary

Air changes per hour — The number of changes of room-air volume per hour due to infiltration air leakage or introduction of ventilation air from outdoors. Based on standard air density of 0.075 lb per cu ft. See *Air recirculations.*

Air-flow rate — The rate of circulation of air in cu ft per min (cfm). Unless otherwise stated, all cfm values are for standard air density of 0.075 lb per cu ft.

Air recirculations per hr — The number of changes of room-air volume per hour due to recirculation of room air only. See *Air changes.*

AST — Average surface temperature: the average value of the surface temperatures of the walls, ceiling and floor, weighted on an area basis.

Balance of room-air temperatures — Uniformity in room-air temperatures between different rooms served by a single room thermostat, as measured at the 30-in. level.

Blower — A centrifugal fan. The warm-air heating industry uses the term to distinguish centrifugal fans from propeller fans.

Blower cycle — One complete cycle of operation from the time the blower begins operation until it begins a second operation, following an off-period.

Bonnet capacity — The heat output of the furnace available at the bonnet, in Btu per hr for a specified air-temperature rise through the furnace.

Bonnet efficiency — The ratio of the bonnet capacity to the heat liberated in the furnace by the burner, also expressed as a percentage. For gas-fired forced-air furnaces approved by the American Gas Association the rated bonnet efficiency is 80 percent.

Breathing-level temperature — Temperature of room air measured at a level 60 in. above floor.

Burner cycle — One complete cycle of operation from the time the burner begins operation until it begins a second operation, following an off-period.

Ceiling-level temperature — Temperature of room air measured at a level 4 in. below ceiling.

Continuous blower operation — A method of blower operation in which continuous operation is approached in average winter weather but intermittent operation is obtained in mild weather.

Convection heating system — In this bulletin the term refers to a conventional forced warm-air heating system, in contrast to a warm-air panel heating system.

Design heat loss — The calculated heat loss for a given space based on outdoor design conditions for the locality. In the text the outdoor design conditions are assumed as −10 F and 15 mph wind velocity.

Duct transmission efficiency — The ratio of the register delivery to bonnet capacity, also expressed as a percentage.

Extended-plenum duct — A trunk duct that is uniform in size along its entire length.

Floor-level temperature — Temperature of room air measured at a level 4 in. above floor.

Fuel consumption — The consumption of fuel per 24 hr. For gas-fired equipment the units are in terms of cu ft of gas per 24 hr.

Fuel input rate — The rate of heat liberation in the furnace by the burner expressed in Btu per hr.

Furnace bonnet — A central plenum, or collecting chamber, located usually above the furnace, in which the heated air is mixed before distribution to the duct system.

Furnace casing — The jacket or enclosure surrounding the furnace. In forced-air furnaces the casing is usually insulated.

Heated basement — Term used when warm air is introduced into the basement through registers in addition to the heat gain from furnace bonnet, furnace casing, ducts, and flue pipe. See *Unheated basement*.

Indoor-outdoor temperature difference — The difference in temperature between indoor air and outdoor air. Large temperature differences denote cold weather, small temperature differences indicate mild weather.

Intermittent blower operation — A term used to designate a method of blower operation in which on-periods and off-periods occur at regular frequencies during normal operation of the system.

Living zone — The space in a room between the floor level and the breathing level.

MRT — Mean radiant temperature at a given location: the mean value of the surface temperatures of the surrounding room surfaces and other objects in the room, taking into account the solid angles which the surfaces make with respect to a unit-receiving surface at the given location.

Panel effect — A heat-transfer effect similar to that obtained from a panel heating system, in which warmed surfaces transmit heat by radiation to cooler surfaces and by convection to cooler air next to the panel surfaces.

Panel heating system — A heating system in which heat is transmitted by both radiation and convection from heated panel surfaces to both air and surrounding surfaces. In this bulletin the term refers to a ceiling-panel installation.

Register delivery — The heat available at the registers, in Btu per hr. This is based on the air-flow rate through the registers and the difference between register-air temperature and the air temperature at the return-air intake.

Sitting-level temperature — Temperature of room air measured at a level 30 in. above floor.

Temperature differential, room-air — The difference in air temperature in a room at two elevations. Usually the sitting level, 30 in. from floor, is considered as the reference level. See *Temperature gradient.*

Temperature gradient, room-air — A represention of air temperatures existing at several levels in a room at one station. See *Temperature differential.*

Thermostat differential setting — An adjustable setting in the room thermostat which governs the degree of fluctuation in room-air temperature at the thermostat.

Total heat input rate — The sum of the fuel input rate and the rate of heat input from lights and other household appliances.

Unheated basement — Term used when no heated air is delivered to basement through registers, although heat regains from the furnace casing, furnace bonnet, ducts, and flue pipe heat the basement air to some extent.

INVESTIGATION WITH HEAT INTRODUCED INTO BASEMENT

II. EQUIPMENT

5. Research Residence No. 2

The Residence is a one-story structure of frame construction with a large amount of glass exposure and with a full basement. The front or north view is shown in Fig. 1a; the rear or south view, in Fig. 1b. The exposed wall section consisted of cedar shingles, 20-lb felt building paper, shiplap sheathing on 2-in.-x-4-in. studdings, 3⅝-in. mineral-wool blanket insulation with vapor barrier attached, and ¼-in. plywood on the interior. The calculated coefficient of heat transmission, U, for this wall section was 0.07 Btu per hr (sq ft) (F). All windows and doors were weatherstripped and were equipped with storm sash and storm doors respectively. Except for one large picture window in the living room, which was fixed in place, the windows were of the horizontal sliding type with storm sash fastened to the window sash. The doors were of conventional wood and glass construction. The heat loss from the structure was calculated by the method given in *Manual 3* (1945 edition)[(2)] which utilizes the coefficients published in the ASHVE *Guide.*[(3)] As recommended in the *Guide*, the infiltration loss was based on a wind velocity of 15 mph and the actual lineal feet of crack around doors and windows.

The heated space consisted of all first-story rooms as well as the entire basement. Table 1 gives a summary of the room dimensions and volumes and of the calculated heat loss for each room. The calculated heat loss from the structure, including the basement, was 60,782 Btu per hr based upon design temperatures of −10 F outdoors and 70 F indoors. This total consisted of 26,285 Btu per hr for the basement and 34,497 Btu per hr for the first story. The total space heated including the basement was 17,212 cu ft.

The Residence included several special features of construction. Open-web steel joists were used in the floor and ceiling so that warm air could be circulated in either joist space to permit investigation of the performance of a heating system using floor or ceiling panels. A 2-in.-thick gypsum plank was used for the subflooring, and asphalt tile for the finish flooring. The ceiling was constructed of pressed cement-asbestos

Fig. 1a. Front View of Warm-Air Heating Research Residence No. 2

Fig. 1b. Rear View of Warm-Air Heating Research Residence No. 2

boards which could be removed for access into the ceiling joist space. Removable plywood boards were used for interior walls to permit access to the vertical stacks in the heating systems.

The Residence was completely furnished (see Fig. 2) and was occupied by a family of two adults. Hence all observations were made under normal living conditions.

6. Heating Systems

To compare the convection and panel heating systems in the Residence, studies were conducted alternately for periods of a few weeks with each system. It was possible, therefore, to obtain performance characteristics of both systems over a wide range of weather conditions.

Two separate duct systems were installed and attached to the same furnace so that the change from the convection to the panel system could

Table 1
Data on Research Residence No. 2

A. Heat Transmission Coefficients, Btu per hr (sq ft) (F)	*U*
Insulated Frame Wall, with 3⅝ in. mineral wool insulation	0.07
Insulated Ceiling, with 3⅝ in. mineral wool insulation	0.07
Outside Doors, equipped with storm doors	0.45
Windows, equipped with storm sash	0.45
Fixed Window in Living Room, double glass	0.45
Basement Wall above grade, 8 in. Haydite block	0.39
Basement Wall below grade, 8 in. Haydite block	0.07
Basement Floor, concrete in contact with ground	0.05

B. Infiltration Factors, cu ft per hr (ft of crack)	*I*
Door, weatherstripped and equipped with storm door	55
Window, weatherstripped	24
Fixed Window in Living Room	14

C. Room Dimensions, Floor Area, Volume, and Calculated Heat Loss

Ceiling heights — first story 8 ft 6 in., basement 8 ft

Room	Room Dimensions	Floor Area, sq ft	Volume, cu ft	Calculated Heat Loss[a] Btu per hr	Calculated Heat Loss[b] Btu per hr
(1)	(2)	(3)	(4)	(5)	(6)
Living Room	21 ft 10 in. x 13 ft 4 in.	292	2 480	11 360	9 510
South Bedroom	13 ft 7 in. x 13 ft 4 in.	156	1 330	5 598	5 598
South Bedroom Closets (2)	4 ft 6 in. x 2 ft 7 in.	23	195	(*)	(*)
Bath	8 ft 0 in. x 5 ft 10 in.	47	395	1 453	1 453
North Bedroom	11 ft 11 in. x 10 ft 0 in.	119	1 010	5 620	5 620
North Bedroom Closet	5 ft 10 in. x 2 ft 8 in.	16	125	(*)	(*)
Hall to Bath	6 ft 7 in. x 5 ft 10 in.	38	325	(*)	(*)
Front Hall	11 ft 6 in. x 5 ft 3 in.	60	514	4 426	2 826
Front Hall Closet	4 ft 0 in. x 2 ft 8 in.	10	90	(*)	(*)
Kitchen-Dinette	19 ft 0 in. x 11 ft 6 in.	219	1 858	6 040	6 040
Total, 1st Story		980	8 322	34 497	31 047
Basement	23 ft 6 in. x 25 ft 3 in. and 28 ft 6 in. x 14 ft 8 in.	1 100	8 890	26 285[c]	21 300[c]
Total		2 080	17 212	60 782	52 347

[a] Heat loss calculations for 1947-48 are based on factors taken from 1945 edition of *Manual 3* of the National Warm Air Heating and Air Conditioning Association.

[b] Heat loss calculations for 1948-49 are based on 1947 edition of *Manual 3* and differ from those shown in col. 5 because of difference in infiltration factors for doors.

[c] Includes stair well from grade level entrance. Values in col. 5 were for 70 F basement-air temperature, and values in col. 6 were for 60 F basement-air temperature. In accordance with the method shown in *Manual 3* of the National Warm Air Heating and Air Conditioning Association a 20 percent reserve capacity was included for the basement.

* Heat loss for these rooms included with larger adjoining rooms.

be made. The following components were common to both systems and were not changed throughout the entire heating season.

The gas-fired forced warm-air furnace having floor dimensions of 2 ft x 3 ft was of the high-boy type with the blower located in the bottom of the unit. The furnace was provided with cast-iron heat exchangers, a burner sized for a rated input of 90,000 Btu per hr,

Fig. 2. Living Room in Warm-Air Heating Research Residence No. 2

and an integrally-mounted, forward-curved, multiblade centrifugal fan having a 12-in. wheel diameter. The humidifier was not operated. Figure 3 shows a view of the furnace located near the center of the basement close to the inside chimney. The chimney was constructed of prefabricated cement-asbestos sectional units. The draft hood was raised 8 in. above the normal position to accommodate instrumentation for measuring the flue gas temperature and the CO_2 content in the flue gas. As discussed later (Section 8) the change in height of the draft hood necessitated a reduction in the flue passage.

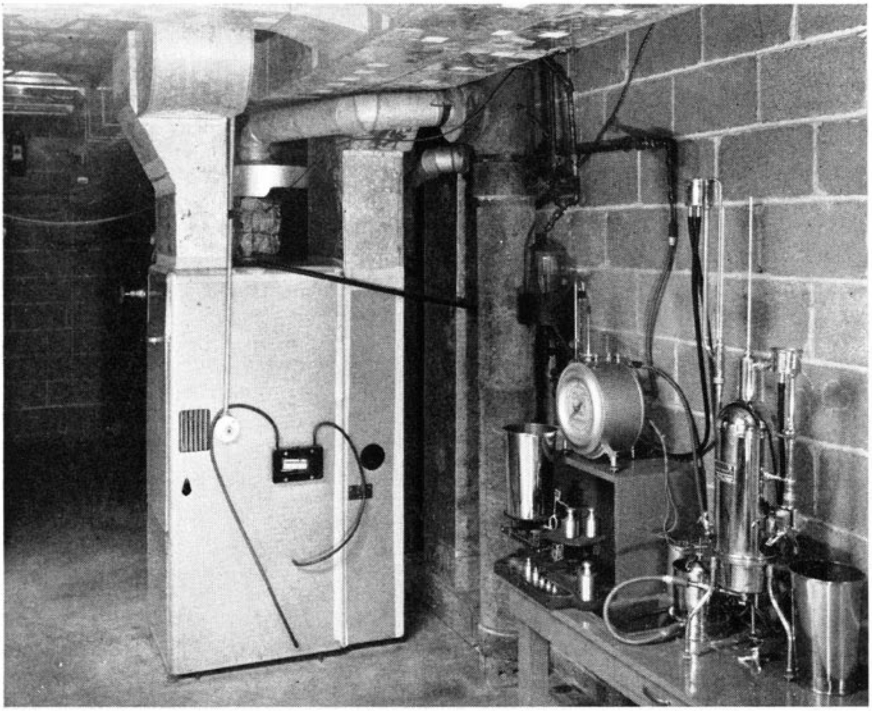


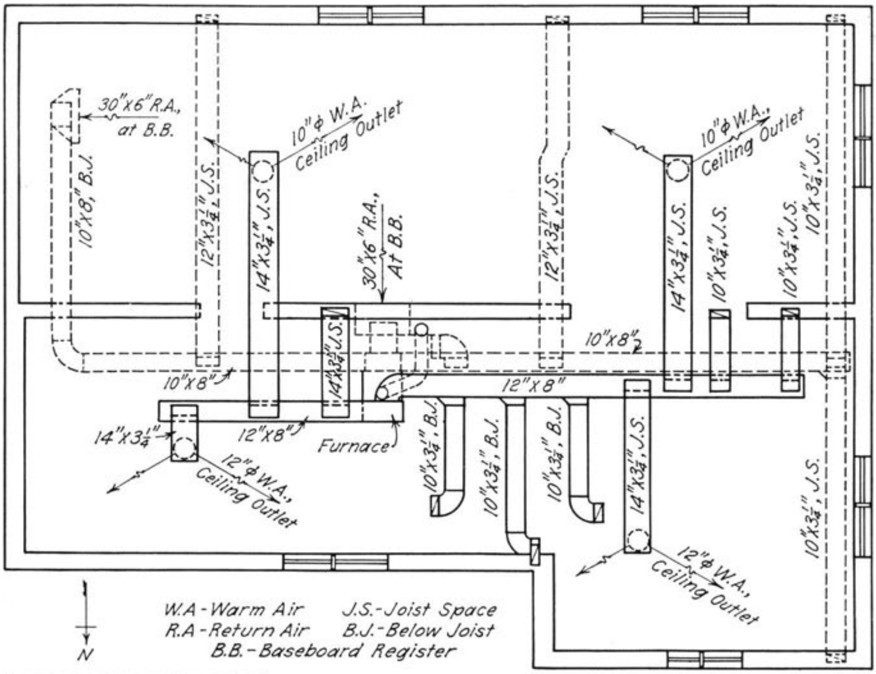
Fig. 3. Gas-Fired Forced-Air Furnace

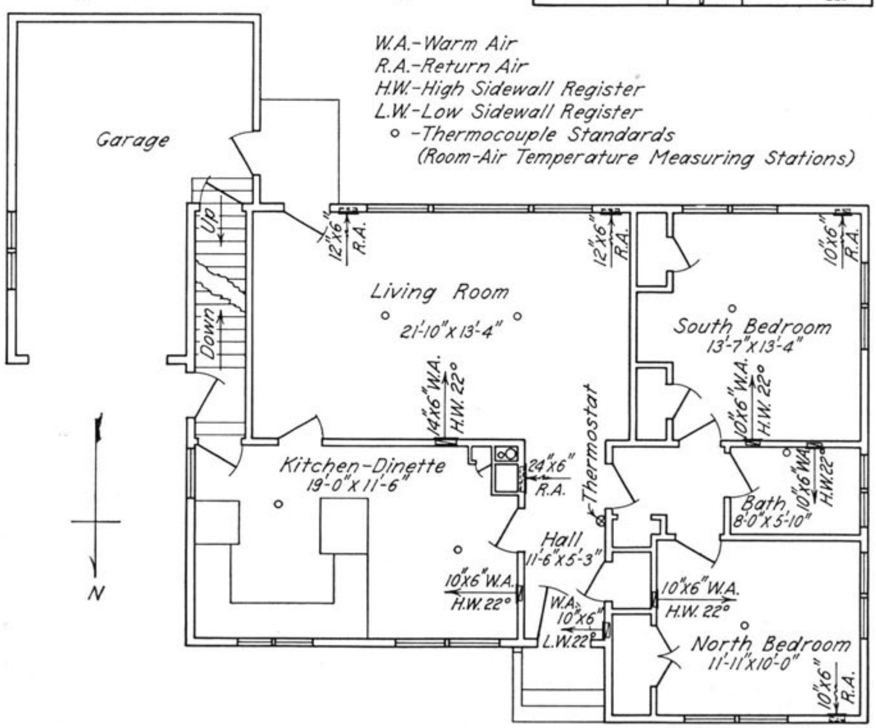
The room thermostat, of the heat-anticipating type, was located in the front hall near the living room at an elevation of 30 in. from the floor. It controlled the operation of the burner in the furnace.

The auxiliary control equipment consisted of a fan switch and a limit control, both located in the furnace bonnet. Operation of the blower was controlled by a fan switch which caused the blower to start or stop operating at the respective cut-in or cut-out points of the fan switch. The limit control in the burner circuit closed the gas valve whenever the bonnet-air temperature exceeded the setting of the control.

(a) Installation A — Convection System

The duct system (Fig. 4a) was of the "extended-plenum" type,[4] having uniformly sized trunk ducts leading from the furnace bonnet toward the east and west ends of the basement. The branch ducts were connected to the top or side of the trunks and were unchanged in size from the trunk take-off fitting to the register stackhead. All registers in the first-story rooms were at the high-sidewall location, 6½ ft from the floor, with the exception of the baseboard register in the front hall near the door. Figure 4b shows the first-story plan and register locations.





Above: Fig. 4a. Basement Plan Showing Extended-Plenum and Branch Ducts for Convection System

Below: Fig. 4b. First-Story Plan Showing Register Locations for Convection System

Ceiling registers of the circular diffuser type were used to distribute warm air into the basement. All return-air intakes were located in the baseboard. The system was designed[5] in accordance with *Manual 7* which, however, did not include equivalent length values for take-off fittings from an extended plenum. Values of 30 and 35 equivalent ft of branch duct were tentatively assigned for the side and top take-off fittings respectively, based upon preliminary investigations conducted in the laboratory.[4] The largest dimension of the trunk duct, as given by the method in *Manual 7*, was used to determine the size of the extended plenum. Preliminary observations made with only two warm-air registers in the south half of the basement indicated that the north half was underheated. To secure a uniform basement-air temperature it was necessary to add two additional warm-air registers in the north half of the basement. No corresponding change was made in the size of the 12-in.-x-8-in. trunk duct.

Since the extended-plenum system was a relatively untried method of air distribution, special attention was given to the possible amount of unbalance observed in the air temperatures maintained in the various rooms and to the steps necessary to overcome the unbalancing. Initially all the dampers in the branch ducts were left wide open and a remarkable uniformity in room-air temperatures was noted. The only corrective measure found to be necessary was to partly close the damper to the dinette, which was located above the furnace. No further adjustments in damper settings were made. The extended-plenum system provided a satisfactory method of air distribution to the various branch ducts.

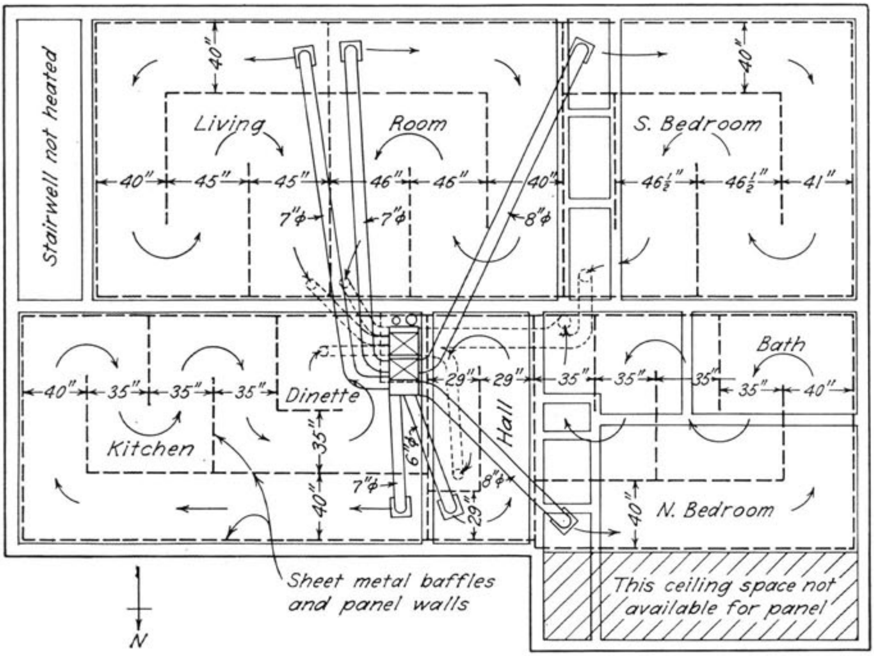
(b) Installation B — Panel System

In panel heating by means of warm air the heated air supplied from the furnace-blower unit is circulated at low velocities behind the panel surfaces. As the circulating air passes through the panel spaces it gives up heat to the panel surfaces which in turn transmit heat both by radiation to the cooler surfaces in the rooms and by convection to the cooler room air. After the circulating air has passed through the panel spaces and has been cooled it returns to the furnace-blower unit for reheating. In the Residence installation the panel surfaces were the individual ceilings of the first-story rooms, and the circulating air was confined entirely to the panel spaces, no air being introduced directly into the rooms. At the time the panel system in the Residence was designed, the only complete design information available was that for the "Panelaire" system.[6, 7] In this proprietary system, the panel space was 3¼ in. in depth, and the air-temperature rise from the inlet to the discharge of the furnace was 55 F. With a few exceptions, as noted below, the design procedure for the Residence installation was the same as that given in reference (7). A later revision[8] of reference (7) was published as *Manual 7-A* by the National Warm Air Heating and Air Conditioning Association.

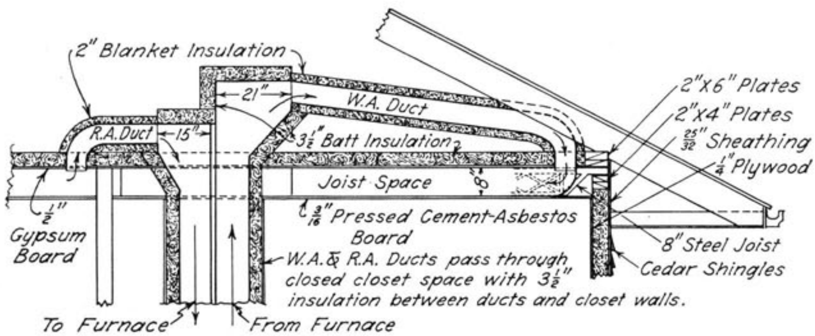
In the case of the Residence installation, since the open-web steel joists were 8 in. in depth and the panel space could be made the same depth with no alterations, it was decided to deviate from the recommendations stated in reference (7) and to use the 8-in. depth. The air velocity through the panel space, therefore, was considerably lower than that recommended in the Panelaire method. The second point of departure from the Panelaire method was in connection with the arbitrary selection of the air-temperature rise. Since all gas-fired furnace-blower units which are tested and rated by the American Gas Association Testing Laboratories provide for a range of air-temperature rises from 70 F to 100 F it was considered desirable to use a temperature rise within this range. Furthermore, since uniformity in ceiling surface temperature could be expected by circulating relatively low-temperature air, the lower value of 70 F temperature rise was selected. This method of operation had the advantage of utilizing the existing furnace-blower unit and did not require the replacement of the blower.

The warm air from the furnace was delivered to an insulated vertical duct and then to a plenum in the attic space. Individual round ducts distributed the warm air from the plenum to six separate panel areas, as shown in Fig. 5a. It may be noted that two separate panel areas served the living room. Each of the six panel spaces was provided with an individual supply of warm air and was sealed to prevent air leakage to or from adjoining panels. Details of the duct system in the attic are shown in Fig. 5b. The air entered the panel spaces near the outside walls of the Residence. By means of sheet-metal baffles, as shown by dotted lines in Fig. 5a, the air was guided over the entire ceiling area of the panels to individual returns near the center of the Residence. After the air had passed through the six panel spaces, it was collected in a common plenum from which it was returned to the furnace-blower unit. A summary of the design data is given in Table 2 (page 23).

The ceiling panel construction consisted of the 8-in. open-web steel joists, $3/16$-in. cement-asbestos boards bolted to the bottom of the joists for the ceiling, and $1/2$-in. gypsum boards placed on top of the joists. Thus, air chambers 8 in. deep were formed by the materials placed on the top and bottom of the joists, all joints being sealed with asbestos paper to prevent air leakage into the rooms or attic space. Mineral-wool batt-type insulation of $3\frac{5}{8}$-in. thickness was placed on top of the gypsum boards, and all ducts in the attic were wrapped with a 2-in. layer of blanket-type insulation.



(a)–Ceiling Panel Arrangement



(b)–Duct Arrangement in Attic for Panel System

Fig. 5. Ceiling Panel Arrangement and Details of Duct Arrangement in Attic for Panel System

Table 2
Data on Design of Panel System

Design based upon "*Panelaire*" *Manual* by H. F. Randolph, International Heater Company, Utica, New York.

Room	Heat Loss, Btu per hr [a b]	Heat Loss, Btu per hr [a c]	Panel Area, sq ft [d]	Calculated Panel Output, Btu per sq ft per hr		Calculated Panel Surface Temperature, F	
				1947-48	1948-49	1947-48	1948-49
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Living Room	13 143	12 507	292[e]	45	43	92	90
South Bedroom and Closets	6 824	7 499	179	38	42	87	90
Bath, Hall to Bath, North Bedroom and Closet	8 055	9 016	165	49	55	94	98
Front Hall	4 711	3 319	60	86	60	117	101
Kitchen-Dinette	7 379	8 284	219	35	40	85	88
Total, 1st Floor	40 102	40 625	915				
Weighted Average				44	45	91	92

[a] Heat loss slightly higher for panel system than that shown in Table 1 because of higher air temperature in ceiling joist space; heat loss through edge of panel included.

[b] Heat loss calculations for 1947-48 are based on factors taken from the 1945 edition of *Manual 3* of the National Warm Air Heating and Air Conditioning Association. These calculations assumed no floor loss to the basement.

[c] Heat loss calculations for 1948-49 are based on factors taken from the 1947 edition of *Manual 3* and differ from those shown for 1947-48 because of difference in infiltration factors for doors. A basement temperature of 60 F was assumed in calculating the floor loss.

[d] Areas based upon distance between panel dividers.

[e] Consists of two panels.

Butterfly dampers were installed close to the warm-air plenum in the attic, in each of the six branch ducts. Initially all dampers were set at a wide-open position and the resulting air temperatures in the rooms were noted. Some temperature unbalance existed; both the living room and the hallway temperatures were higher than average while the north bedroom temperature was lower. To improve the balance it was found necessary to deliver more air to the north bedroom, since the panel area was only about 50 percent of the entire ceiling area. Suitable adjustments in damper positions were made at an outdoor temperature of about 35 F, and once the dampers were set no further adjustments were made.

7. Instrumentation

To measure temperatures, approximately 200 thermocouples of 24-gauge copper-constantan wire were installed. Thermocouples were placed at four different levels on standards located near the centers of each of the first-story rooms as shown in Fig. 4b, and at three stations in the basement. Thermocouples were also installed on the ceiling and floor surfaces, in the attic, in the duct systems, and at other desired points inside and outside the Residence. All thermocouples were connected to two switchboards on the instrument panel in the basement. Each of the switchboards was connected to an indicating potentiometer shown in Fig. 6. By means of a twelve-point recording potentiometer it was possible to obtain a continuous record of the temperature at any twelve of the 200 stations.

Resistance thermometers having greater sensitivity than thermocouples were installed at the sitting level on the standards in five of the

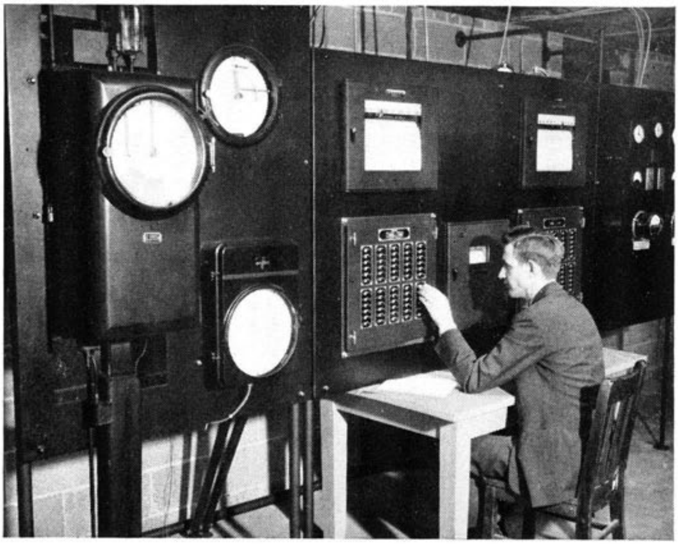


Fig. 6. Instrument Panel in Research Residence No. 2

first-story rooms and at one station in the basement. The resistance thermometers were connected to a six-point recording potentiometer which provided a continuous record of the temperatures.

The temperature of the air leaving the bonnet was measured in the trunk ducts by means of two thermocouple grids, each consisting of six thermocouples connected in parallel. These grids were so located that they were not affected by direct radiation from the heat exchangers in the furnace. The flue-gas temperature was measured by means of a recording thermometer. An instrument located in the living room was used to record the relative humidity of the air. Continuous records were obtained of the draft at the base of the chimney with a draft recorder, and of the CO_2 content in the flue gases below the draft hood with a recorder. The electrical inputs to the burner and to the blower motor were measured by watt-hour meters reading directly to 10 watt-hours. Self-starting electric clocks were connected across the circuits for the burner and for the blower motor in order to obtain the total time of operation of each. The fuel input to the furnace was determined from one gas meter; the gas used for cooking and water heating was determined from a separate meter.

The air-flow rate for each system was determined by means of a vane-type anemometer, installed in the return-air duct at the furnace. This anemometer was calibrated in position by the method discussed in Appendix A.

III. PROCEDURE

8. Preliminary Statement

For both systems the thermostat setting was maintained at 72 F, with the temperature differential adjusted to the minimum setting so that frequent burner operations were obtained. The resulting room-air temperatures at the sitting level were maintained at approximately 72 F throughout the 24 hr constituting a test period. All doors between rooms were open unless otherwise stated.

The fuel used was natural gas having a calorific value of 1000 Btu per cu ft. The fuel input rate was reduced from the normal rated value for the furnace of 90,000 Btu per hr to the desired value of 76,000 Btu per hr by adjusting the flow rate at the meter. No trouble was experienced from flame failure. This input rate of 76,000 Btu per hr for the convection system was determined by dividing the total heat loss of the Residence by the assumed bonnet efficiency of the furnace, $\frac{60{,}782}{0.80} = 76{,}000$ Btu per hr. Since the basement was to be heated, the duct heat loss from bonnet to register was neglected in this calculation.

Because of the greater loss upward and outward from the panel space, the calculated heat loss for the panel system was found to be 5605 Btu per hr greater than for the convection system. Furthermore, the duct transmission efficiency was considered to be lower than that assumed for the convection system, since the branch ducts were located in an unheated attic. However, to compare the two systems directly, the fuel input rate of 76,000 Btu per hr was also used for the panel system.

In order to install instrumentation at the flue outlet for measuring the temperature and the CO_2 content of the flue gas, it was necessary to raise the draft hood in the manner shown in Fig. 3. Raising the draft hood and reducing the fuel input rate to 76,000 Btu per hr resulted in an excessive air supply to the burner. This condition was corrected by restricting the flue passage until a CO_2 content in the flue gas of 8.5 percent was obtained.

Either periodic or continuous records were made of all significant temperatures, such as those for room air at the floor level, the sitting level, the breathing level, and the ceiling level; basement air at the same levels; floor and ceiling surfaces; air in the duct systems; outdoor air; and attic air. Complete daily records were made of the operating time, the number of cycles of operation, and the electrical consumptions of the gas valve, the blower motor and the total for the Residence. Daily observations were made of the gas consumed in the furnace and the gas used for household purposes.

9. Experimental Conditions

Four main series of investigations were conducted in order to study and compare the performances of the two systems in accordance with the objectives stated in Section 3. The operating conditions for each series are given in Table 3. The studies of the convection system have been designated as series A and those of the panel system as series B. Both series A-1 and B-1 refer to "continuous blower operation" — with low settings of the fan switch; series A-2 and B-2 refer to "intermittent blower operation" — with relatively high settings of the fan switch. For the convection system a flow rate of 565 cfm was used, corresponding to 2.0 air recirculations per hour. This flow rate resulted in an air temperature rise of 100 F between the furnace inlet and discharge when steady-state conditions were maintained and when new filters of the throw-away type were in place. For the panel system the flow rate was 795 cfm, corresponding to a temperature rise of 70 F with no filters in place.

Table 3
Experimental Conditions for 1947-48 Season

Series	Type of System	cfm	Fan-Switch Settings		Limit-Control Settings		Room Thermostat Setting, F	Period of Observation
			Cut-in, F	Cut-out, F	Cut-out, F	Cut-in, F		
A-1*	Convection	565	100	80	200	185	72	Oct. 28–Nov. 11 Jan. 13–Jan. 16 Jan. 24–Jan. 25 Mar. 30–Apr. 9
A-2	Convection	565	150	125	200	185	72	Dec. 30–Jan. 7 Jan. 11–Jan. 12 Jan. 17–Jan. 23 Apr. 10–Apr. 18
A-3	Convection	565	100	80	200	185	72 (6 a.m.–10 p.m.) 62 (10 p.m.– 6 a.m.)	Jan. 8–Jan. 10
B-1	Panel	795	100	80	170	155	72	Nov. 13–Dec. 7 Jan. 26–Jan. 31 Mar. 10–Mar. 29
B-2	Panel	795	140	110	170	155	72	Dec. 8–Dec. 21 Feb. 1–Mar. 1
B-3	Panel	795	140	110	170	155	72 (6 a.m.–10 p.m.) 62 (10 p.m.– 6 a.m.)	Dec. 22–Dec. 23
B-4	Panel	795	140	110	170	155	72 (6 a.m.–10 p.m.) 67 (10 p.m.– 6 a.m.)	Dec. 24–Dec. 26
B-5	Panel	795	100	80	170	155	72 (6 a.m.–10 p.m.) 67 (10 p.m.– 6 a.m.)	Dec. 27–Dec. 29

* The method of operation used in series A-1 conforms to the principle of circulating air as continuously as possible as outlined in *Manual 6* of the National Warm Air Heating and Air Conditioning Association.

IV. ROOM-AIR TEMPERATURES, MEAN SURFACE TEMPERATURES, AND AVERAGE RELATIVE HUMIDITIES

10. Preliminary Statement

Comfort depends on a large number of factors such as air temperatures, surface temperatures, relative humidity, and air movement as well as a number of more subjective items such as odor, noise, and dust content. Since it was not possible to evaluate all these items, emphasis was placed on two predominant factors — air temperatures and relative humidity. The study of air temperatures was devoted primarily to conditions obtained in the living zone between the floor level and the breathing level. The air temperature at the ceiling level is of little significance in determining comfort conditions in homes having average ceiling heights of 8 ft or greater. However, in order to minimize the heat loss through the upper exposed walls and through the ceiling, a low air temperature above the breathing level is desirable.

11. Room-Air Temperature Variations During Cycling of Burner and Blower

As indicated in Fig. 7, the cyclical variation in living-zone temperatures in the south bedroom was small for each of the four main series. This small variation was characteristic of the conditions which were maintained in all of the rooms on the first story under all weather conditions.

With the panel system (series B-1 and B-2) the air temperatures at the ceiling level were uniform as compared with those experienced with the convection system (series A-1 and A-2). However, the same type of temperature fluctuations occurred within the panel as were experienced at the ceiling level with the convection system. Obviously, in the case of the panel system the ceiling produced a stabilizing effect, so that the air temperatures immediately below the ceiling remained practically constant. In spite of larger variations of air temperatures at the ceiling level with the convection system, the variations in the living zone were similar for the two systems; they amounted to a maximum of about 0.5 F during one complete cycle of burner operation.

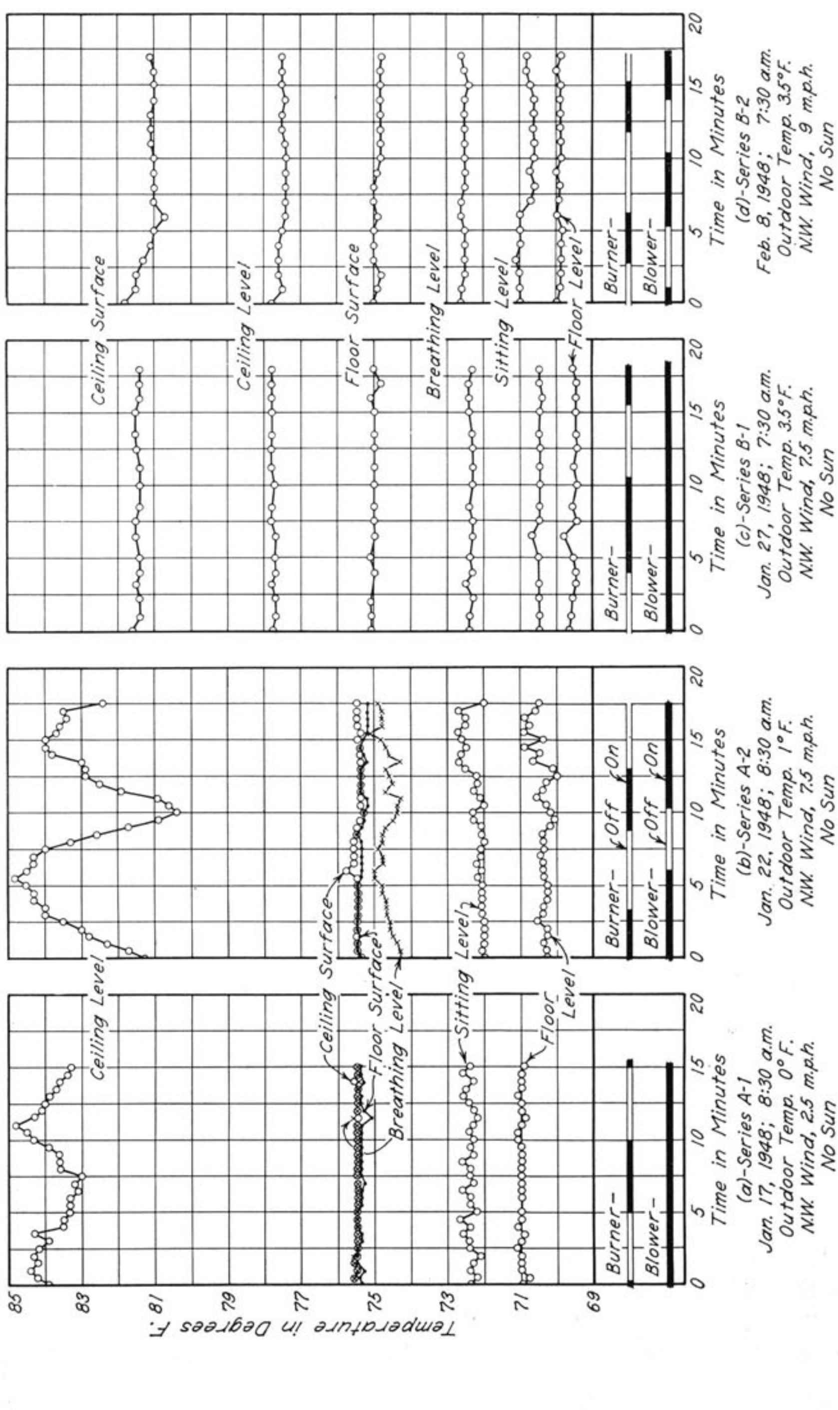


Fig. 7. Room-Air Temperature Variations in South Bedroom During Cycling of Burner and Blower

In all these studies it was found that as a result of frequent blower operations and resultant short off-periods the intermittent blower operation (series A-2 and series B-2) produced nearly the same constant temperatures in the living zone as continuous blower operation (series A-1 and series B-1). Since frequency of blower operation depended on frequency of burner operation, which in turn was governed by the thermostat sensitivity, it was concluded that (1) the heat-anticipating characteristics and the differential setting of the room thermostat were very important in obtaining close control of air temperatures in the living zone; (2) the conventional thermostat performed equally well for both the panel system and the convection system; and (3) the rate of response for the panel system was sufficient to follow normal changes in outdoor temperature, and no evidence of thermal lag with resultant overheating was noted. It should be emphasized, however, that the type of ceiling panel in use provided relatively small heat storage capacity.

12. Room-Air Temperature Differences Between Rooms

The differences in room-air temperatures from one room to another may vary with the outdoor temperature and the wind velocity even though the heating system has been balanced to produce the minimum temperature difference during average winter weather with moderate wind velocities. Variations in temperature difference are caused to a large extent by the infiltration of outdoor air, the amount of infiltration depending on the tightness of house construction. Table 4 shows the maximum temperature differences between rooms for four weather conditions. The maximum differences were slightly greater for the panel system, because the lowest temperature of all rooms was observed in the north bedroom. From Fig. 5 it may be noted that the usable panel area in the north bedroom was only one-half the ceiling area. Because of the undersized panel the air temperatures in the north bedroom were approximately 1 F lower than the average temperature for the Residence. Any attempts to reduce the difference would have required extreme adjustments in the air-flow rate to each of the panels, and since the north bedroom air temperature was only 1 F lower, these adjustments were not considered feasible. If the minimum temperatures for the north bedroom were not considered, the maximum differences in temperatures between rooms corresponded closely with those experienced with the convection system. Table 4 shows that the variation in temperature difference caused by changes in wind velocity and infiltration was of the same order of magnitude for each system and was within tolerable limits. The type of blower operation — continuous or intermittent — had no significant effect upon the temperature difference between rooms.

Table 4
Maximum Temperature Differences Between Room at Sitting Level (Heated Basement)

All values were obtained from the continuous records made during night-time operation when no influence from solar heat gain was experienced.

Outdoor Temperature, F	Wind Velocity, mph	Max. Diff. in Temp. at Sitting Level Between Rooms, Caused by Balance of Systems and Weather Conditions, F	Increase in Max. Difference at Sitting Level Caused by Increased Wind Velocity, F
		Convection System	
31.0	0	0.9	...
33.0	12.0	0.9	0
7.0	1.5	1.0	...
8.0	12.0	2.1	1.1
		Panel System	
31.0	0	1.9	...
36.0	12.0	2.4	0.5
6.0	0	2.2	...
3.5	8.0	3.5	1.3

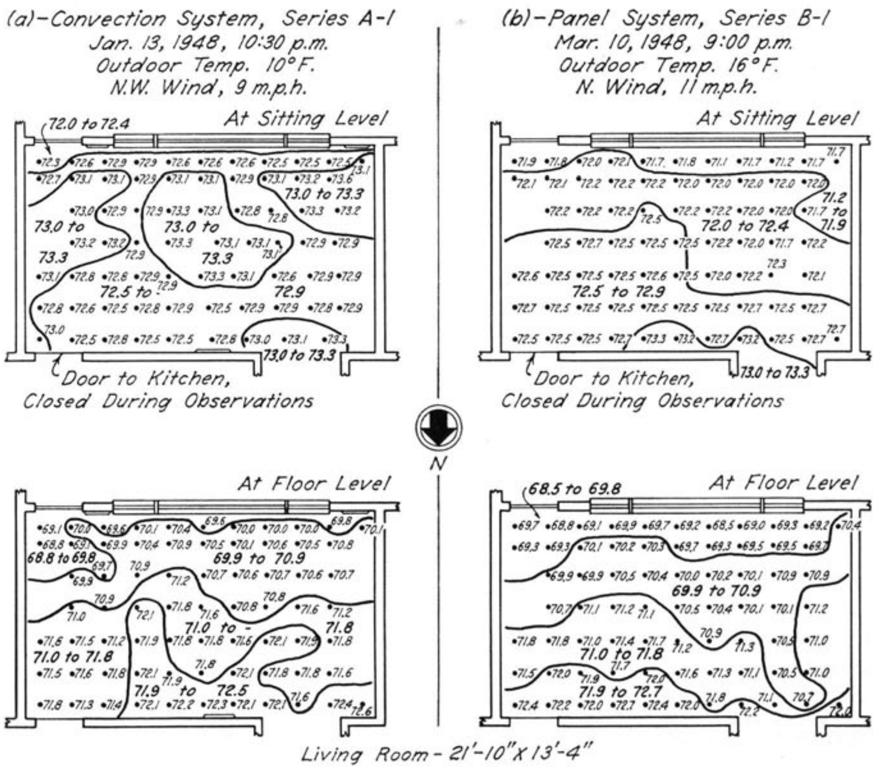


Fig. 8. Isothermal Contours in Living Room (Heated Basement)

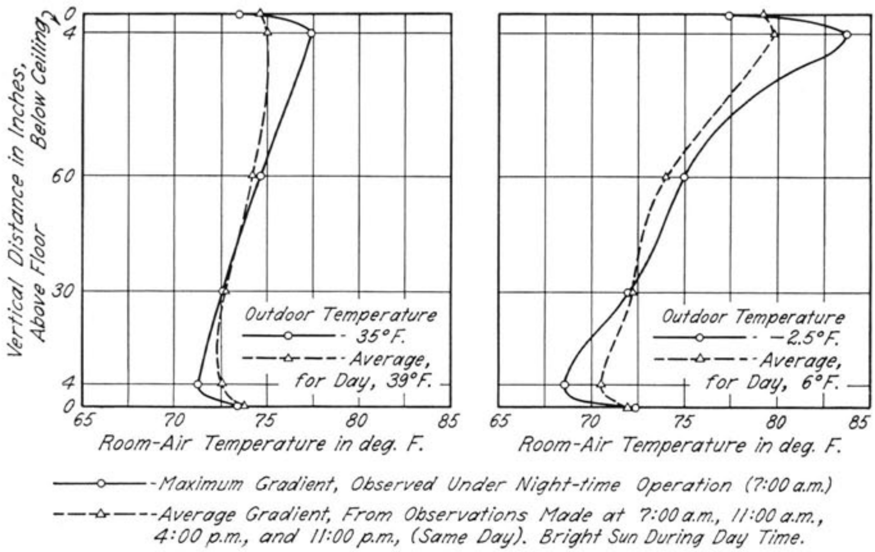


Fig. 9. Room-Air Temperature Gradients for Living Room with Convection System (Series A-1)

13. Air Temperatures in Living Room

Observations were made in the living room of room-air temperatures in two horizontal planes by means of portable thermocouples. The living room was selected for this study because the large amount of glass area in the exposed wall should have resulted in a relatively large difference in temperatures in the horizontal plane. Figure 8 shows isothermal contours based on temperatures obtained at 69 different points in the living room at both the sitting level and the floor level. Observations were made under night-time operation for each system, to eliminate the influence of solar heat gain.

As shown in Fig. 8a, the convection system produced air temperatures at the sitting level which varied from a maximum of 73.3 F at the inside wall to a minimum of 72.0 F near the outside door. Figure 8b shows that at this same level the variation for the panel system was from 73.3 F at the inside wall to 71.2 F near the outside wall and large glass area. Figure 8a also shows that at the floor level the variation from maximum to minimum for the convection system was from 72.5 F to 68.8 F, whereas (Fig. 8b) the variation for the panel system was from 72.7 F to 68.5 F. The difference between the two systems was slight. However, the low temperature zone near the exposed wall covered a larger room area in the case of the panel system than with the convection system.

14. Room-Air Temperature Differentials

The room-air temperatures obtained in a vertical plane at any given location of the thermocouple standard in a room can best be depicted by a graph in which the air temperatures observed at four elevations in the room are plotted against the vertical distances above the floor level, as shown by the typical example in Fig. 9. These curves, designated as "room-air temperature gradients," apply to the central location of the standard and are considered to be representative of the temperature gradients in a large part of the given room. In general a gradient which most nearly approaches a vertical line is considered to represent the most favorable condition of room-air temperatures. This is particularly true in the living zone between the floor level and the breathing level. In the zone between breathing level and ceiling level any deviation from the vertical line, if not excessive, is of little importance as far as comfort conditions are concerned.

The slope of any room-air temperature gradient depends largely on the outdoor weather conditions and on the operation of the heating system as it functions to meet the demands imposed by these conditions. In addition the gradient is affected by extraneous heat gains such as those from solar heat effects, cooking, bathing, etc., none of which are produced by the system. These extraneous heat gains had the least effect on the temperature observations made in the early morning hours after a long period of night-time operation. Previously, in Research Residence No. 1, the room-air temperatures observed at 7 a.m., 11 a.m., 4 p.m., and 11 p.m. were averaged for the day, and the average values were plotted against the indoor-outdoor temperature difference for the same day. However, in the smaller Research Residence No. 2, for which solar heat effects were much more pronounced, it was observed that the room-air temperature gradients were affected by the solar heat effects to such an extent that the differences in gradients for two methods of plant operation were largely obscured. For example, the gradients observed at 11 a.m. and 4 p.m. were most favorable from the standpoint of comfort conditions, particularly when solar heat effects were experienced, and much more favorable than those obtained at 7 a.m. or even at 11 p.m.

The two gradients on the left side of Fig. 9 show the difference in gradients obtained on a given day by two methods of depicting the data. The broken-line gradient represents an average of four readings made in the living room at 7 a.m., 11 a.m., 4 p.m., and 11 p.m. on a given average day. The solid-line gradient represents only the 7 a.m. readings; it is not as favorable as the other gradient. For example, the temperature differential between the floor level and the breathing level was only 1.7 F for the average of four readings but was 3.3 F for the 7 a.m. reading.

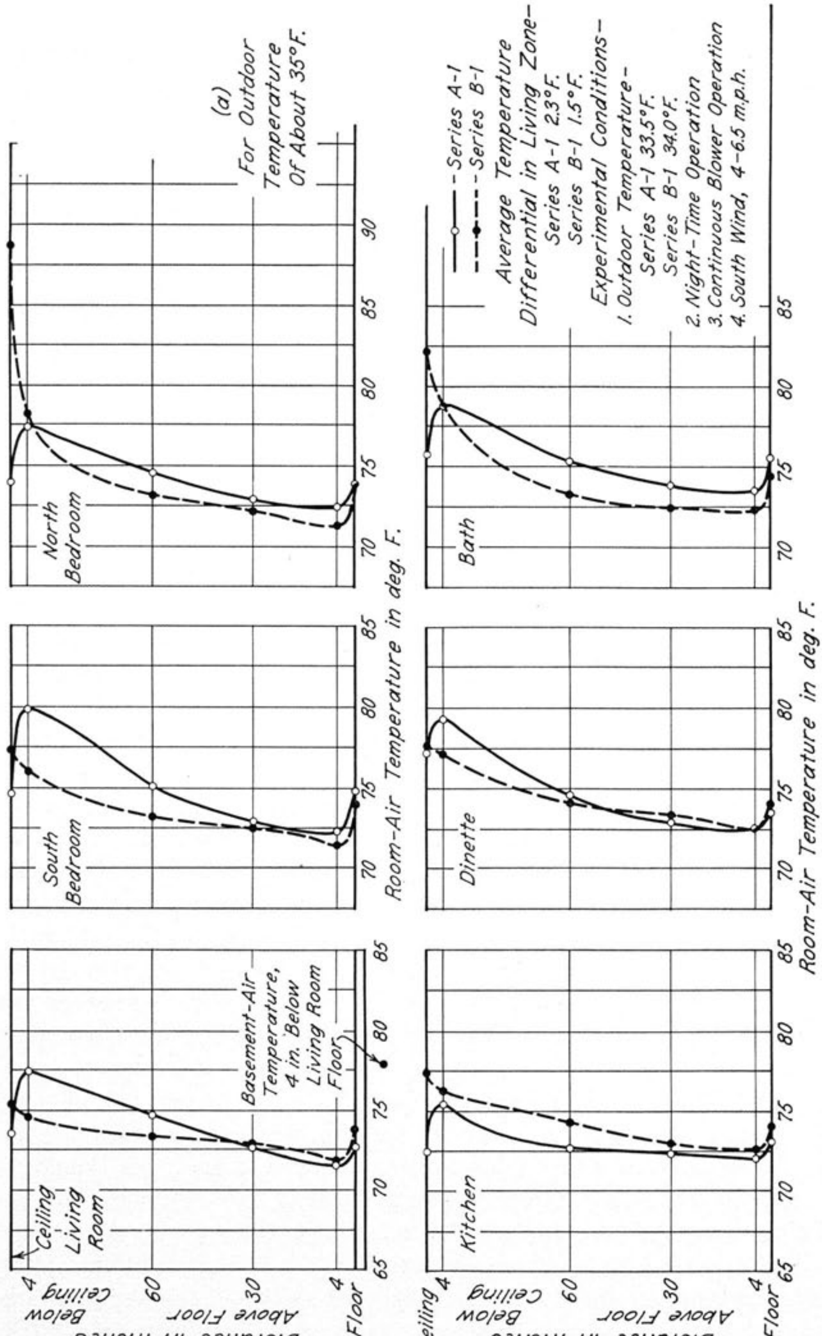


Fig. 10a. Room-Air Temperature Gradients for an Outdoor Temperature of About 35 F (Heated Basement)

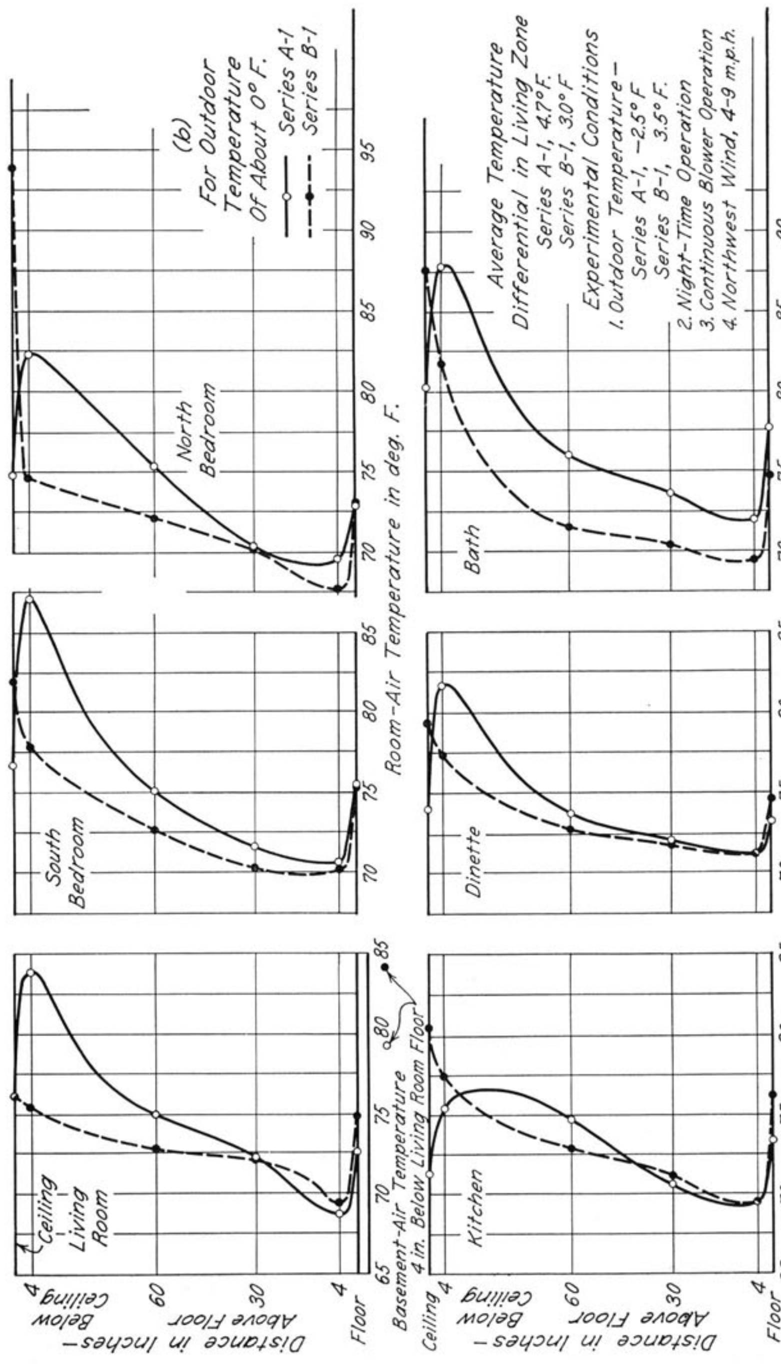


Fig. 10b. Room-Air Temperature Gradients for an Outdoor Temperature of About 0 F (Heated Basement)

Similar conditions are shown by the gradients on the right side of Fig. 9 for a colder day. In this example the temperature differential from the floor level to the breathing level was 3.0 F for the average of four readings but 6.6 F for the 7 a.m. readings.

It is true that from the standpoint of the occupants the average of four readings gives a better representation of temperature gradients during the day than does the single reading at 7 a.m. However, from the viewpoint of comparing the performances of different heating systems, the observations made at 7 a.m. best depict the plant performance unaffected by solar-heat gain or living operations. Hence, in spite of the fact that plant performance was not shown in its more favorable aspects, the practice was adopted of including only the 7 a.m. temperature gradients.

Figures 10a and 10b show typical room-air temperature gradients for each of the first-story rooms with both the convection and panel systems. The curves in Fig. 10a represent data for average winter weather; those in Fig. 10b are for an outdoor temperature of about 0 F. A marked similarity in temperature gradients existed for the two systems, the differentials being greater for both systems in colder weather. However, the temperature differential in the living zone of each room was slightly greater for the convection system than for the panel system.

As stated in Section 9, the room-thermostat setting was maintained at 72 F during the four main series. However, as may be seen in Figs. 10a and 10b the average room-air temperatures were about 0.8 F lower for the panel system than for the convection system. This difference was found to exist throughout the season, but was not noticeable as far as the comfort sensations of the occupants were concerned. Apparently the thermostat was influenced by radiation from the ceiling panel, causing the thermostat to be satisfied before the air temperatures reached the same value as those obtained with the convection system.

With both systems the floor-surface temperatures were greater than the air temperatures at the floor level. These higher floor-surface temperatures were undoubtedly caused in the main by heat transmission through the floor from the heated air immediately below the floor and to some extent by radiation from the warm ceiling surface.

Observations were made of room-air temperatures at 7 a.m. without regard to the operating cycle of the burner and blower. These periodic observations (Fig. 7) gave a satisfactory index of room-air temperatures, since the temperature variations caused by the cycling of the burner and blower were insignificant. A comparison of room-air temperature differentials representing an average for all rooms of the Residence and covering a wide range of weather conditions is shown in Fig. 11. The temperature at the sitting level was used as the reference temperature. The difference between this reference temperature and those at the floor, breathing, and ceiling levels was plotted against the indoor-outdoor temperature difference. Like the room-air temperature gradients, these temperature differentials were plotted only for the 7 a.m. observations, and did not include those made during the hours of 11 a.m., 4 p.m., and 11 p.m. Hence, the differentials shown in Fig. 11 can be considered as maximum values that provide the best index of the comparative performances of the heating systems. On this basis the temperature differentials observed with the convection system were greater than those with the panel system. For example, at an outdoor temperature of 35 F

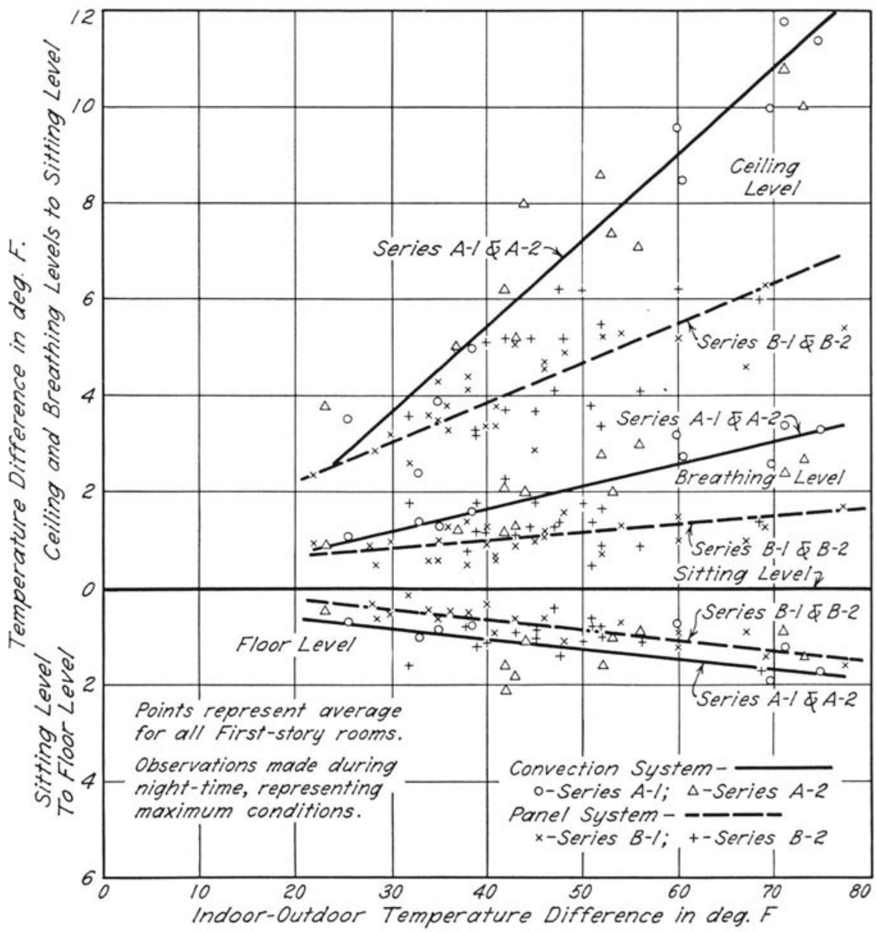


Fig. 11. Room-Air Temperature Differentials (Heated Basement)

the average temperature differential in the living zone was about 2.5 F for the convection system as compared with 1.5 F for the panel system. At an outdoor temperature of 0 F this differential was about 4.5 F for the convection system and 3.0 F for the panel system.

From the observations made of room-air temperature gradients and room-air temperature differentials it may be concluded that the panel system produced a slightly better temperature condition in the living zone than did the convection system.

15. Average Surface Temperatures in Living Room

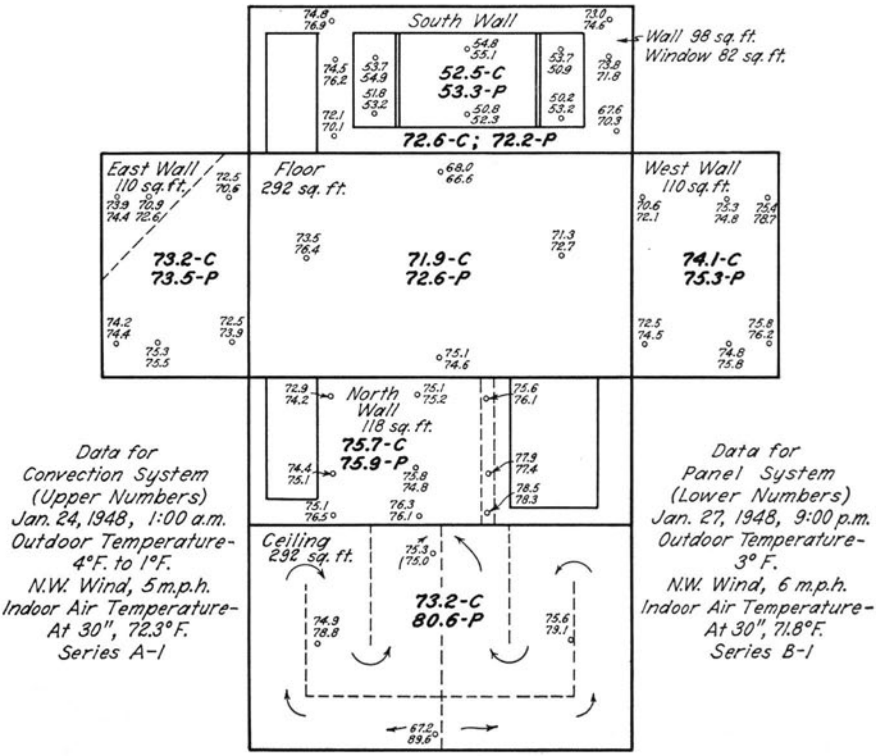
As defined by the ASHVE *Guide*,[9] "The rate of heat loss by radiation depends upon the exposed surface area of the body, and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls or other objects. This latter temperature is called the Mean Radiant Temperature (MRT).

"Because these two types of heat loss (radiation and convection) supplement each other, a required rate of total heat loss can result either from a relatively low air temperature and a relatively high MRT, or vice versa."

It might be expected that the MRT for the panel system would be higher than that for the convection system. If such were actually the case, it should be possible to maintain lower air temperatures for the panel system than for the convection system under comparable conditions of comfort. The determination of MRT requires the use of an instrument such as the thermo-integrator which evaluates the MRT at the location of the instrument. The surface of the instrument is considered to be exposed to a mean radiant temperature, which in turn is dependent on the temperature of each of the surrounding surfaces and the included solid angles that the surfaces make with respect to the surface area of the instrument. Theoretically, therefore, an infinite number of values of MRT are obtainable for the infinite number of positions at which the instrument can be located in the room. The determination of MRT for a specific location was made during the 1948-49 heating season, and the results are discussed in detail in Section 31.

For the 1947-48 season, however, when the thermo-integrator was not available, it was considered that some approximation of the MRT could be made by determining the average surface temperature (AST) in the living room. It should be noted that the AST does not depend on the position of the occupant in the room, but is merely a weighted average temperature of all the surfaces which enclose the room — ceiling, floor, walls, and windows. Detailed studies of surface temperatures in the living room were conducted on two nights during which the outdoor temperature was about 3 F. The temperatures at 41 separate points on the six interior surfaces comprising the living room boundary were measured by means of thermocouples attached to the surfaces. The average temperature for each separate room surface as well as the AST for the room is shown in Fig. 12. The AST was evaluated by dividing the sum of products of surface areas and temperatures by the sum of the areas.

A comparison of the AST values obtained with the two systems is shown at the bottom of Fig. 12. Considering all six surfaces of the room, the AST for the panel system was only 1.7 F higher than that for the convection system. Except for the ceiling surface all corresponding surface temperatures were within about 1 F of each other for the two



Average Surface Temperatures—
a.—Excluding Ceiling Surface— A-1 71.6°F., B-1 70.4°F.
b.—Including Ceiling Surface— A-1 71.5°F., B-1 73.2°F.

Fig. 12. Average Surface Temperatures in Living Room

systems. In other words, the small difference in AST was due almost entirely to the difference in ceiling-surface temperatures. That the AST difference was not greater than 1.7 F may be attributed partly to the fact that the actual ceiling-surface temperature (80.6 F) for the panel system was considerably lower than calculations (89 F) indicated. The causes of the difference were the heat regains from the chimney and through the floor, as well as from electrical appliances and occupants, all of which tended to reduce the required output of the ceiling panel.

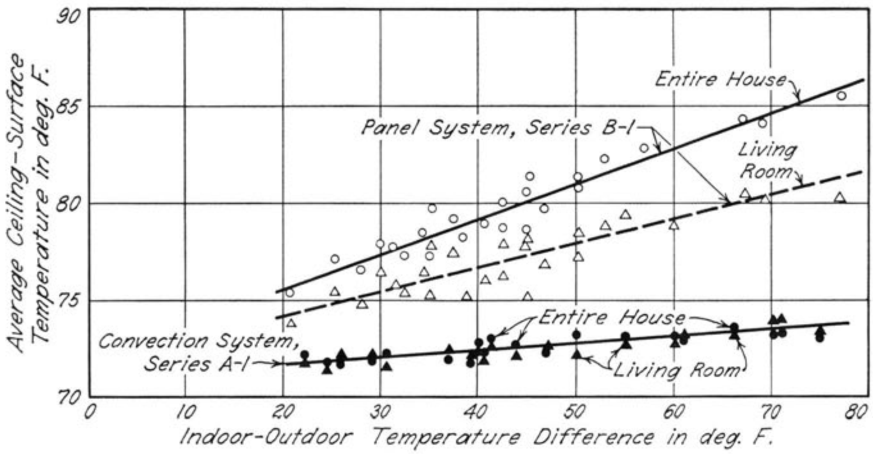


Fig. 13. Average Ceiling-Surface Temperatures

The small difference in AST may also be attributed to the fact that the convection system operated to a certain extent as a panel system. As noted before, the addition of heat to the basement resulted in the formation of a layer of warm air just below the first floor. This produced a floor-panel effect and resulted in floor-surface temperatures which were somewhat higher than might have been expected. This same panel effect also existed at the north wall, where in the case of the convection system the chimney and the warm-air stack behind the wall created a warm-wall surface.

It is true that the slight difference in the AST shown in Fig. 12 was not necessarily representative of all of the remaining rooms in the Residence. Figure 13 shows that in the case of the panel system the average ceiling-surface temperature for the entire house was about 4 F higher than that for the living room at an indoor-outdoor temperature

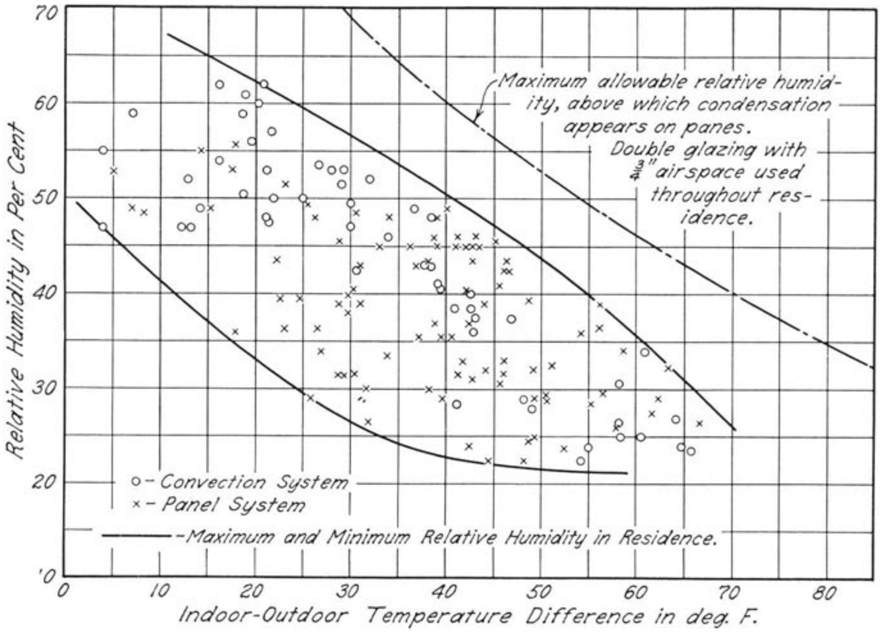


Fig. 14. Average Daily Relative Humidities

difference of 70 F. However, with the convection system the ceiling surface temperature for the entire house was practically the same as that for the living room. Since the ceiling area comprised about one-fourth of the total area of the room surfaces, this difference of 4 F in ceiling-surface temperatures would result in an increase in AST for the entire house of only about 1 F. Furthermore, as noted in Section 31, the difference between air temperature and AST was found to be of the same order of magnitude as that between the air temperature and MRT. In any case, the AST values for the convection and panel systems were not markedly different; they were slightly in favor of the panel system.

16. Average Daily Relative Humidities

To determine the basic level of prevailing relative humidity in the Residence, it was deemed desirable to operate the plant without the addition of moisture to the circulating air. Hence the humidifier in the furnace was not operated. The average daily relative humidities experienced with both systems over a wide range of indoor-outdoor temperature differences are shown in Fig. 14. Considerable variation was obtained between the maximum and minimum relative humidities for a given temperature difference. This large variation may be accounted for partly by differences in moisture input as a result of washing, cooking, etc., but to a larger extent by the method used in plotting the data. The average value of the daily relative humidity was plotted against the average value of the indoor-outdoor temperature difference for the same day. As long as the outdoor temperature was fairly constant, the relative humidity was also constant; but when the outdoor temperature changed sharply, a slower change in relative humidity occurred and a large scattering of the plotted points resulted. Thus the indoor relative humidities were governed to a large extent by the weather conditions prevailing on the preceding day. As a result, wide variations in maximum and minimum values of relative humidity were obtained for any given indoor-outdoor temperature difference.

In this phase of the study there was no discernible difference between the two systems. The minimum relative humidity experienced was about 22 percent in cold weather, which was lower than that usually recommended. A curve representing the maximum allowable relative humidities above which condensation appears on the panes for double-glazed windows[10] is also included in Fig. 14. No condensation was experienced with the double-glazed windows in the Residence.

17. Summary of Room Temperatures

From the observations made of the room-air temperatures, average surface temperatures, and relative humidities, the performances of the panel and convection systems were remarkably alike. However, the panel system produced a wider zone of minimum air temperatures near the exposed wall in the living room than did the convection system, even though the range of temperature from maximum to minimum was about the same. On the other hand, the room-air temperature differentials in the living zone were less for the panel system than for the convection system.

No difficulty was experienced with either system with the automatic temperature control, even though a conventional room thermostat was used for both the convection and panel systems. No temperature overruns or thermal lags were experienced with the panel system, probably because of the small heat storage capacity of the distributing system. Hence, from the standpoint of factors affecting comfort, both systems produced satisfactory results.

V. PLANT PERFORMANCE

18. Performance of Burner and Furnace

The operating characteristics of the burner and blower not only influence the comfort produced but also give an indication of the cost of operating the system. Figures 15a and 15b show performance curves for the burner and furnace over a wide range of indoor-outdoor temperature differences. A broken line is used to indicate intermittent blower operation (series A-2 and B-2); a solid line, the condition in which the blower operated practically continuously in cold weather (series A-1 and B-1).

A comparison of the fuel consumption curves in Figs. 15a and 15b indicates that the consumption was approximately 20 percent higher for the panel system than for the convection system. A substantial part of this increase could be accounted for by the heat loss from the attic ducts. These branch ducts connecting the attic plenum to the individual panels were made sufficiently long to permit the determination of the air-flow rate to each panel by means of pitot-tube traverses. Had these branches been made shorter the duct heat loss could have been reduced materially. Or had the system been so designed that the individual panels were supplied by means of warm-air stacks located within the inside walls, the duct heat loss could have been made available as a heat regain. The rest of the increase in fuel consumption could be accounted for by the fact that the calculated heat loss for the panel system (66,387 Btu per hr) was about 9 percent higher than that for the convection system (60,782 Btu per hr). This difference of 5605 Btu per hr was due to the higher air temperature in the ceiling joist space, which in turn resulted in a larger heat loss above the panel and through the edges of the panel. A minimum fuel consumption for a ceiling panel system requires adequate insulation of all exposed duct work as well as of the upper surface and edges of the panel area.

To determine whether the increase in fuel consumption was due to a stratification of the heated air in the panel space, a study of the temperature gradients in the panel space was made, as discussed in Appendix B. If a marked stratification had occurred and a layer of high-temperature air had existed immediately below the upper surface of the panel space, the heat loss from the panel space to the attic would have been larger than if no stratification occurred. The evidence presented in Appendix B shows that no such stratification did occur.

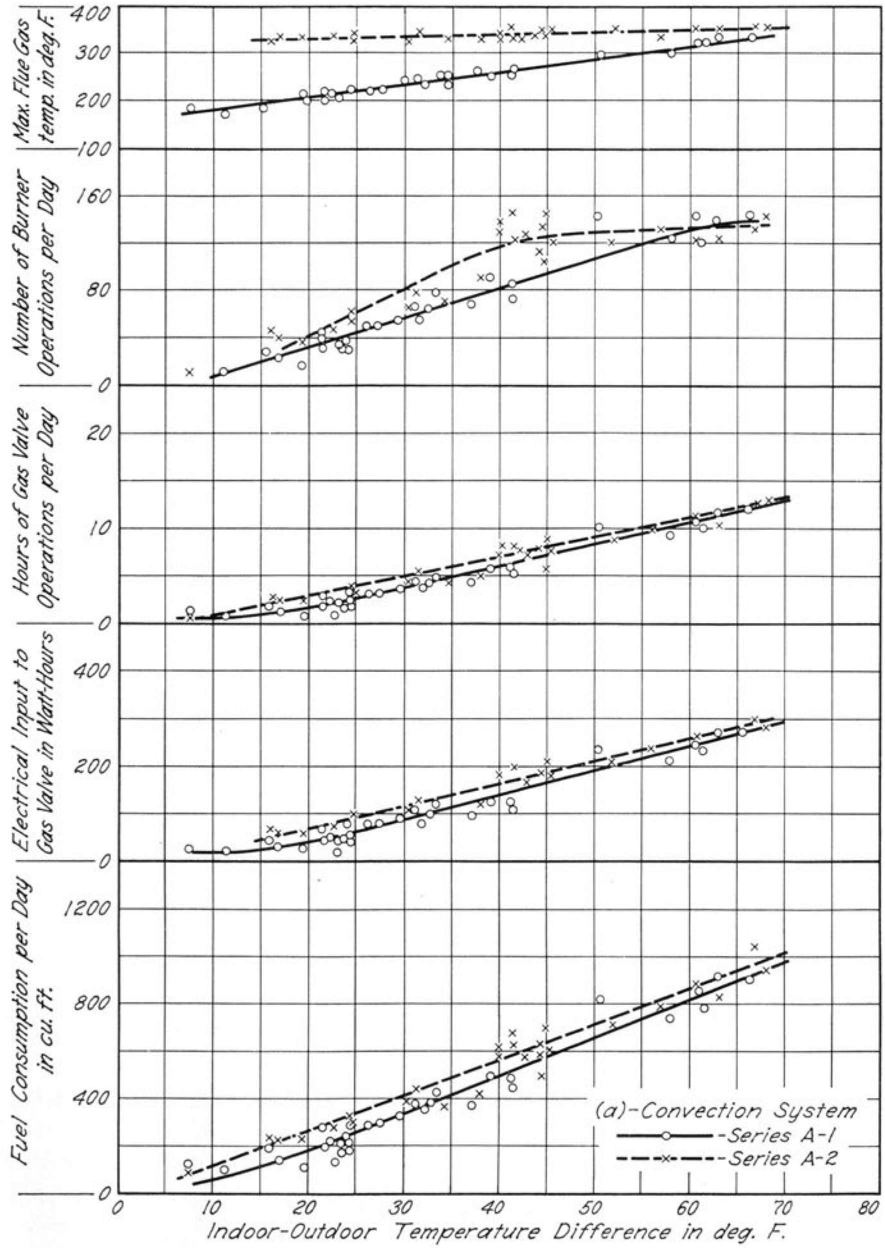


Fig. 15a. Performance of Burner and Furnace with Convection System (Heated Basement)

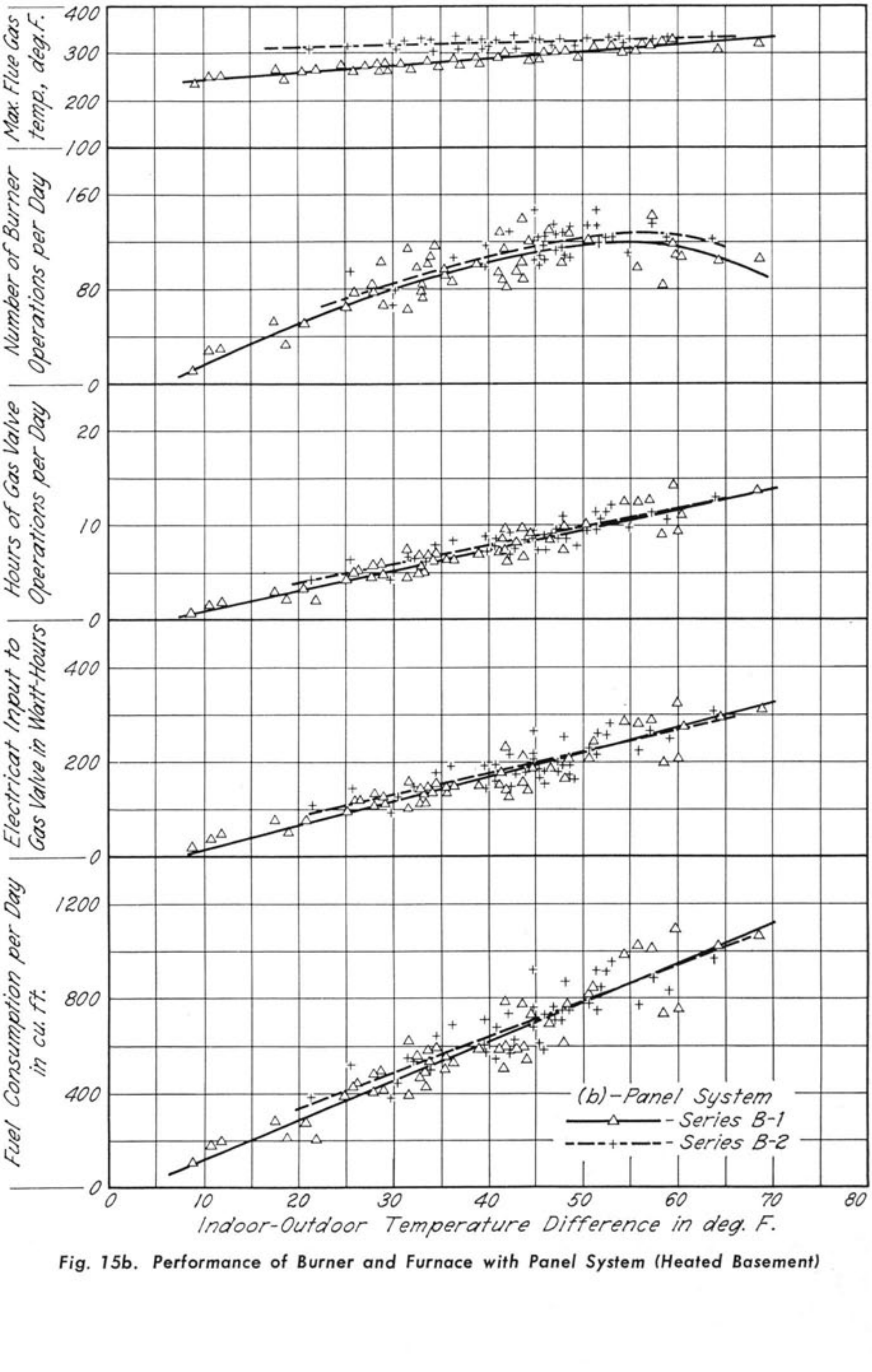


Fig. 15b. Performance of Burner and Furnace with Panel System (Heated Basement)

The difference in fuel consumption between the two systems might have been reduced slightly had a shallower panel depth been used. If a panel of 3¼-in. depth were used instead of the 8-in. depth, the air velocity in the panel would be about 2½ times greater, causing a reduction of thickness of the surface films on the inside of the panel. The resulting decrease in resistance to heat transfer through the films would have little effect on the over-all resistance to heat flow upward from the panel, but would cause a more rapid rate of heat transfer from the panel downward into the rooms. This in turn would result in a lower mean air temperature within the panel, and a lower rate of heat transfer upward and outward from the panel.

Figures 15a and 15b also afford a means for comparing the fuel consumptions for the two ranges of fan-switch settings used. The high settings of the fan switch (series A-2 for convection and B-2 for panel) resulted in intermittent operation of the blower over the entire range of weather conditions. On the other hand, the low settings of the fan switch (series A-1 for convection and B-1 for panel) resulted in intermittent operation for weather warmer than about 30 F and continuous operation for weather colder than 30 F. This low setting of the fan switch was in accordance with the "continuous air circulation" principle advocated by the National Warm Air Heating and Air Conditioning Association.[11] The method of control used in series A-1 and B-1 has been referred to in this bulletin as "continuous blower operation." With both the convection and panel systems, intermittent operation of the blower resulted in a fuel consumption which was about 10 percent greater than that for the continuous blower operation. As is shown at the top of Figs. 15a and 15b, intermittent blower operation was accompanied by higher flue-gas temperatures and consequently by a larger flue-gas loss. With the higher settings of the fan switch the blower did not begin to operate and transfer heat from the furnace until the flue-gas temperature had reached a higher level. The difference in fuel consumption decreased in cold weather, when the high setting of the fan switch also produced practically continuous blower operation.

The curves for both the electrical input to the gas valve and the total time of operation exhibited the same trends as did the curves for fuel consumption. Since in no case did the burner operate more than 15 hr per day during 0 F weather, it may be concluded that the fuel input rate of 76,000 Btu per hr was more than sufficient. A further discussion of fuel input rates and heat losses is presented in Sections 38 and 39.

19. Performance of Blower

Figures 16a and 16b show the performance curves for the blower over a range of indoor-outdoor temperature differences. Higher electrical inputs were required with continuous operation than with intermittent operation of the blower except in extremely cold weather, when the values coincided. To a large extent, however, the increase in electrical cost was offset by the reduction in fuel cost.

The electrical input for the panel system was higher than that for the convection system for similar methods of operation. With the panel system the blower not only operated longer, but since the air-flow rate and blower speed were greater, the electrical input to the blower motor was also greater. From the curves shown at the top of Figs. 16a and 16b it may be noted that when the blower operated continuously the bonnet temperatures increased as the heating demand became larger. However, when the blower operated intermittently, the bonnet temperatures remained relatively constant. The average bonnet temperatures were far less than the design values of 165 F for the convection system and 135 F for the panel system. This discrepancy has been attributed to the fact that the heat loss from the Residence was offset by heat delivery, not only from the heating system proper but from heat regain (Section 39).

20. Response Characteristics with Night Set-Back of Thermostat

A study was made with a reduced night setting of the room thermostat to determine the response and lag characteristics of both systems. The outdoor temperature was between 25 F and 30 F for all series of observations. With series A-3, in which the thermostat setting was reduced 10 F between the hours of 10 p.m. and 6 a.m., the time required for the convection system to recover the 10 F in the morning was approximately 2½ hr. With series B-3, however, in which the same thermostat settings were used, the time required for the panel system to recover the 10 F in the morning was approximately 5 hr, about twice as long as the time required for the convection system. Series B-4 and B-5 with the panel system were conducted with a reduced night setting of the thermostat of only 5 F between the hours of 10 p.m. and 6 a.m., and for these series the time of recovery was approximately 2½ hr, or the same as that required in series A-3. These studies indicated that it was not practical to reduce the thermostat setting more than 5 F when operating the panel system, since a larger set-back required an excessively long period of recovery.

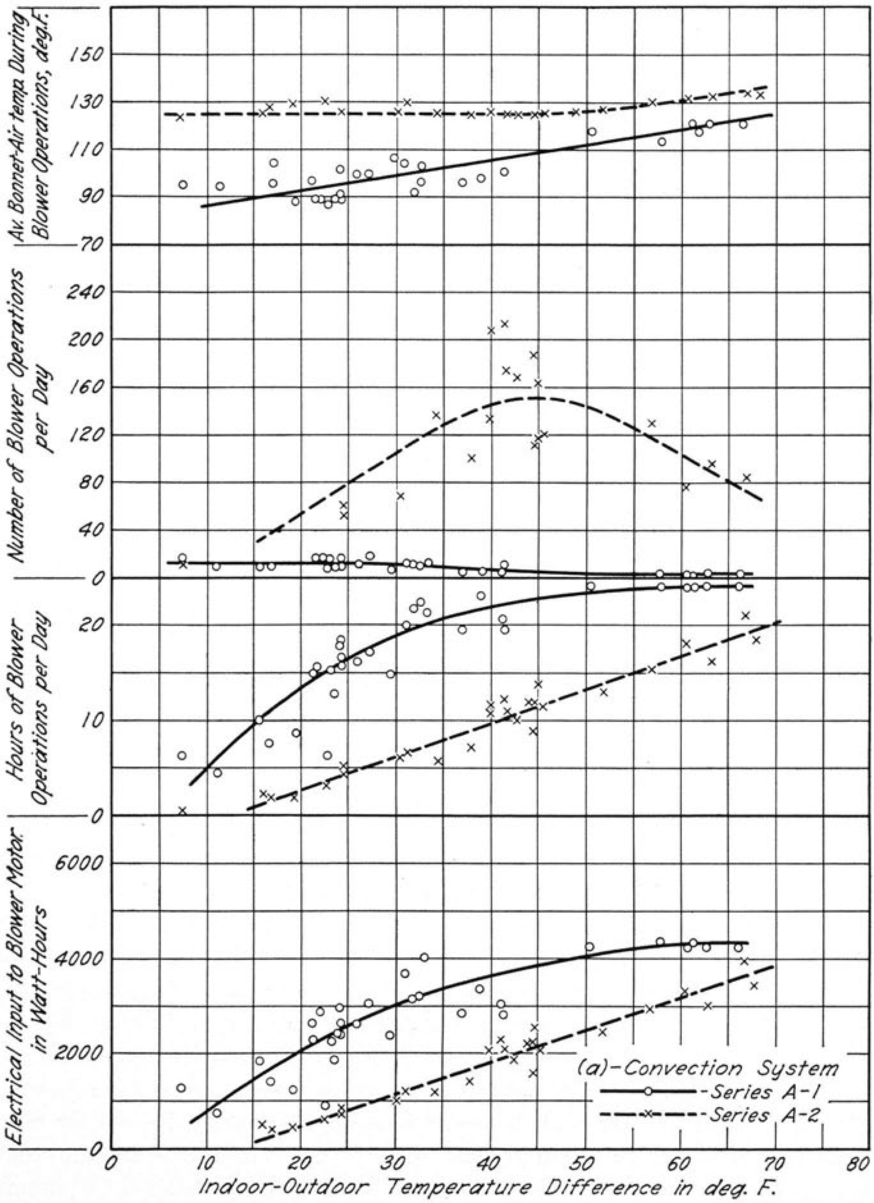


Fig. 16a. Performance of Blower with Convection System (Heated Basement)

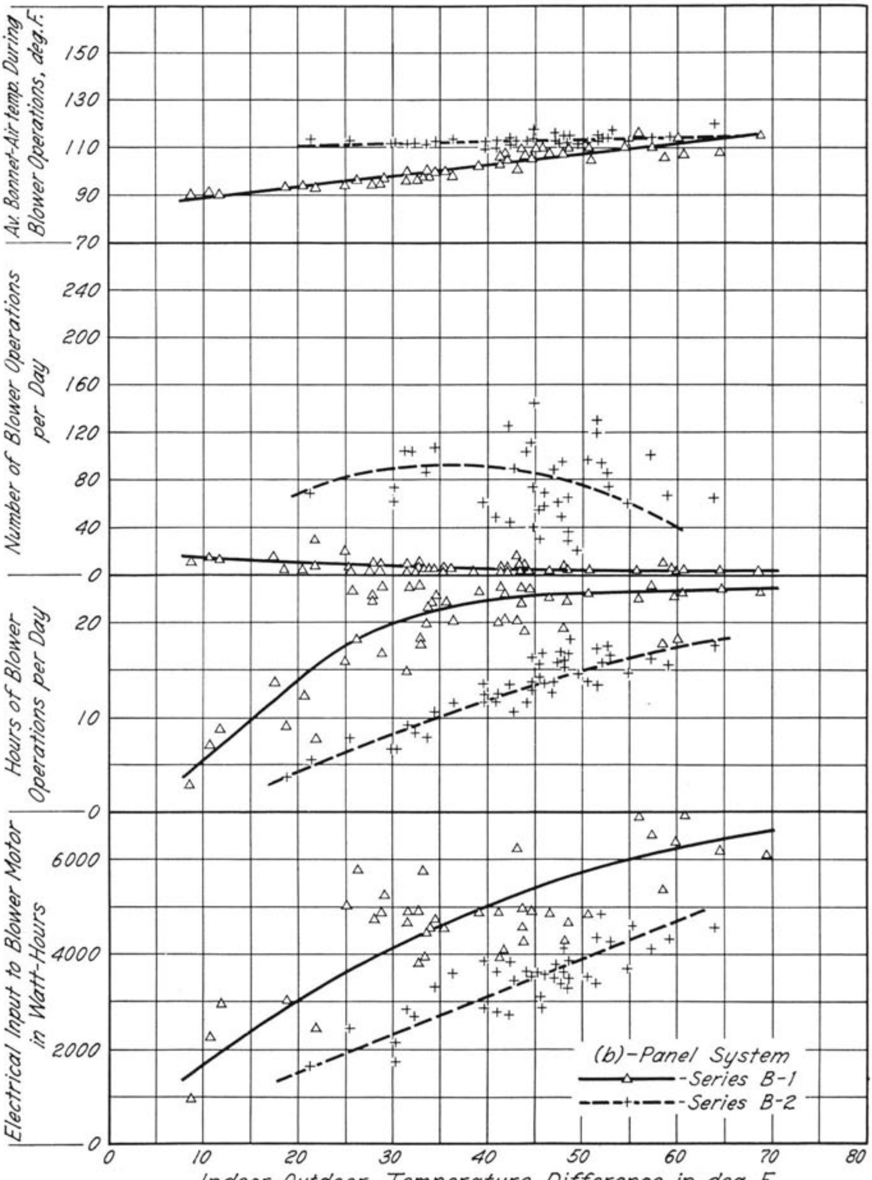


Fig. 16b. Performance of Blower with Panel System (Heated Basement)

21. Summary of Plant Performance

A slightly higher fuel consumption can be expected with a ceiling panel system than with a convection system when there is an open and vented attic above the panel. Though the results obtained showed an increase in fuel consumption of about 20 percent for the panel system, a large part of this increase was due to the construction of the panel system in which long branch ducts in the attic were used. With a system using short branch ducts, or possibly wall stacks, the difference in fuel consumption should have been considerably less. The investigation indicated the need for adequately insulating all duct work located in cold spaces where heat cannot be regained into the heated space. Adequate insulation of the top side of the panel space in the attic as well as of the exposed edges of the panel space was also shown to be essential. Slightly higher electrical inputs were required with the panel system, primarily because of the greater air-flow rate. The response characteristics of the systems with constant thermostat setting were the same, but when the thermostat setting was reduced 10 F for an 8-hr night-time period, the panel system required about twice as much time to recover the 10 F as did the convection system.

Intermittent blower operation resulted in higher fuel consumption but lower electrical consumption than did continuous blower operation.

INVESTIGATION WITH NO HEAT INTRODUCED INTO BASEMENT

VI. EQUIPMENT AND PROCEDURE

22. Preliminary Statement

As mentioned previously (page 40), the heated air introduced into the basement through the ceiling registers during the 1947-48 heating season produced a floor-panel effect that was observed with both the convection and panel systems. This effect could not be considered typical of systems in which no heat is introduced into the basement. Therefore, in order to obtain performance data for the systems operating with no heat introduced into the basement the investigation was continued during the 1948-49 heating season.

In addition a study of the effect of the differential setting of the room thermostat upon the performance of the convection system was considered to be desirable. Hence two additional series of studies were conducted during 1948-49 with the differential setting of the room thermostat adjusted to the maximum value.

23. Equipment

The description of the Residence in Section 5 applies equally well to the 1948-49 season. The calculated heat loss of the first-story rooms was 31,047 Btu per hr based on the heat transfer coefficients given in the 1947 edition[12] of *Manual 3*. This differs from the 34,497 Btu per hr heat loss for the first story, which was based on the 1945 edition[2] and was used in 1947-48. The difference was due to revisions in infiltration factors for doors. Both of these heat loss values were determined in the usual manner, since no account was taken of the possible heat transfer through the floor surface to or from the basement. An itemized tabulation of pertinent data is shown in Table 1. Although no heat was introduced into the basement through registers, the presence of the furnace, flue pipe, and duct system would contribute some heat regain; accordingly it was expected that the heat transfer through the floor to the basement would be relatively small. Since the same family of two adults occupied the Residence during both heating seasons, all studies were conducted under similar living conditions.

The heating systems used in 1948-49 were the same as in the previous season except that the ducts leading to the four basement registers were sealed at the trunk. It was realized that with this duct arrangement the trunk duct was slightly oversized; however, no changes were made. Plans for the convection and panel systems are shown in Figs. 4a and 5a respectively. Details for the design of the panel system as operated in 1948-49 are shown in Table 2, columns 6 and 8. In the case of the panel system, the possible source of heat regain was confined to the furnace casing and flue pipe, since practically none of the duct system was located in the basement. Hence it was assumed that the basement-air temperature would be about 60 F and that the heat transfer through the floor surface would be negligible. Upon this basis, the total heat loss for the first story was found to be 40,625 Btu per hr (Table 2, column 3). The slight differences shown in the itemized values listed in columns 2 and 3 of Table 2 indicate that the existing branch duct sizes of the panel system were adequate.

Except for additional thermocouples installed on the underside of the subfloor and in the attic ducts, the instrumentation for the Residence in 1948-49 was essentially the same as in the previous year. For the purpose of studying the mean radiant temperature (MRT), as discussed in Section 15, a thermo-integrator was located in the living room and measurements were made with both the panel and the convection systems.

24. Procedure

For both the panel and the convection systems the thermostat setting was maintained at 72 F. All doors between rooms were open unless otherwise stated. The fuel input rate to the furnace was reduced from the rated value of 90,000 Btu per hr to the desired value of 46,000 Btu per hr. This input rate was determined by dividing the heat loss of the first-story rooms only by the assumed bonnet efficiency of the furnace, 0.80, and the assumed duct transmission efficiency, 0.85. That is, $46{,}000 = \dfrac{31{,}047}{(0.80)\,(0.85)}$. This method of selecting the fuel input rate is commonly used for convection heating systems in which the furnace and ductwork are located in the basement and in which no provisions are made for introducing warm air into the basement to maintain a 70-F temperature. Furthermore, since any floor loss from the first-story rooms to the basement is usually neglected, it was also omitted from the total of 31,047 Btu per hr given above. No trouble from flame failure was experienced with the reduced fuel input rate.

Because of the greater loss from the top side and edges of the panel and the loss into the basement, the heat loss for the panel system was calculated to be 9578 Btu per hr greater than that for the convection system. This indicates that the fuel input rate should have been greater for the panel system, and of the order of 59,700 Btu per hr. However, as in the 1947-48 investigation, in order to make a direct comparison of all factors influenced by the fuel input rate, the same rate was used for both systems. Since the actual input rate of 46,000 Btu per hr was considerably less than the required indicated rate, some difficulty could be expected in maintaining the Residence at 72 F under design weather conditions.

Table 5
Experimental Conditions for 1948-49 Season

Series	Type of System	cfm	Fan-Switch Settings		Limit-Control Settings		Room Thermostat Setting, F	Room Thermostat Differential Setting	Period of Observation
			Cut-in, F	Cut-out, F	Cut-out, F	Cut-in, F			
A-11*	Convection	340	100	80	200	185	72	Minimum	Nov. 5–Nov. 17 Dec. 18–Dec. 27 Jan. 19–Jan. 26
A-12	Convection	340	150	125	200	185	72	Minimum	Dec. 28–Jan. 5 Jan. 27–Feb. 1
A-13	Convection	340	100	80	200	185	72	Maximum	Jan. 15–Jan. 18 Feb. 24–Mar. 12 Mar. 28–Apr. 3
A-14	Convection	340	150	125	200	185	72	Maximum	Jan. 6–Jan. 14 Mar. 12–Mar. 27
B-11	Panel	485	100	80	170	155	72	Minimum	Nov. 19–Dec. 1 Dec. 13–Dec. 17 Feb. 2–Feb. 23
B-12	Panel	485	150	125	170	155	72	Minimum	Apr. 4–Apr. 9 Dec. 2–Dec. 12

* The method of operation used in series A-11 conforms to the principle of circulating air as continuously as possible, as outlined in *Manual 6* of the National Warm Air Heating and Air Conditioning Association

As discussed in Section 8, the raised draft hood (Fig. 3) and the reduced fuel input rate resulted in an excessive air supply to the burner. The restriction in the flue passage which was made in 1947-48 in order to reduce the amount of excess air and to obtain a CO_2 content in the flue gas of 8.5 percent was left unchanged, with the result that the CO_2 content in the flue gas was reduced to about 5.5 percent. This same flue restriction was used because it was not possible to raise the CO_2 content appreciably without closing the secondary air openings into the furnace, a practice which is not recommended.

As shown in Table 5, the four series conducted were similar to those conducted in the previous year. For series A-11 and A-12 with the convection system, the air-flow rate was 340 cfm as determined from the fuel input rate of 46,000 Btu per hr and a 100-F temperature rise through the furnace. This flow rate corresponded to 2.5 air recirculations per hr, and compares with the 565 cfm circulated for the corresponding series A-1 and A-2 of the previous year. Similarly for series B-11 and B-12 with the panel system, the air-flow rate was reduced to 485 cfm as determined from the same fuel input rate and a 70-F temperature rise through the furnace. This compares with the 795 cfm used with the corresponding series B-1 and B-2 of the previous year. Two additional series were conducted to determine the effect obtained with the maximum setting of the thermostat differential. These series, A-13 and A-14, correspond otherwise to A-11 and A-12 respectively.

In general, the periodic and continuous records made of all significant temperatures and other data were the same as those made during the previous season. The relative humidity data, obtained with no heat introduced into the basement and with the humidifier not in operation, were identical with those shown in Fig. 14.

Although no heated air was introduced into the basement by means of registers, the heat regain from the furnace casing, furnace bonnet, flue pipe, and duct system was of such magnitude that the basement air temperature did not drop below 57 F even in the coldest weather. Hence, strictly speaking, the basement was not unheated for either method of operation, and heat loss occurred from the basement to the outdoors. This sizable basement heat loss was ignored in the heat loss calculations, but it did exist and (see Section 38) should have been taken into account in considering the total heat input rate to the Residence. For the sake of brevity, the term "unheated basement" used hereafter in the bulletin refers to the condition where no heat was introduced directly into the basement through registers.

VII. ROOM-AIR TEMPERATURES, CEILING- AND FLOOR-SURFACE TEMPERATURES, AND MEAN RADIANT TEMPERATURES

25. Room-Air Temperature Variations During Cycling of Burner and Blower

The cyclical variations of the temperatures in the living zone were small for each of the four main series with the unheated basement, and were of the same magnitude as those experienced during the 1947-48 season with the heated basement. These favorable results should have been anticipated, since no change was made in the heating system or the control settings. Even with intermittent blower operation, the room-air temperatures remained essentially constant because the blower operated frequently and the off-periods were of short duration. The response of the panel system to sudden changes in outdoor temperature was similar to that experienced with the heated basement.

26. Room-Air Temperature Differences Between Rooms

The temperature differences between rooms were small whether the basement was heated or unheated and whether the blower operation was continuous or intermittent. Neither the reduction in air-flow rate nor the blocking of the ducts to the basement affected the temperaure balance of the rooms, and therefore no adjustment of dampers was required. The maximum temperature differences observed between rooms were 1.4 F for the convection system and 1.9 F for the panel system. These differences, both of which were smaller than those observed with the heated basement, may be accounted for by the fact that high winds and low outdoor temperatures did not occur simultaneously in the studies made when the basement was unheated.

27. Air Temperatures in Living Room

The room-air temperatures measured in the two horizontal planes in the living room are shown in Fig. 17. The maximum differences in temperature at the sitting level were 2.2 F for the convection system and 0.8 F for the panel system. At the floor level the maximum difference was 2.9 F for the convection system and 4.1 F for the panel system. The results indicated a slightly greater uniformity in temperature for the convection system, since the cooler area near the exposed walls was smaller. The results were essentially the same as those with the heated basement.

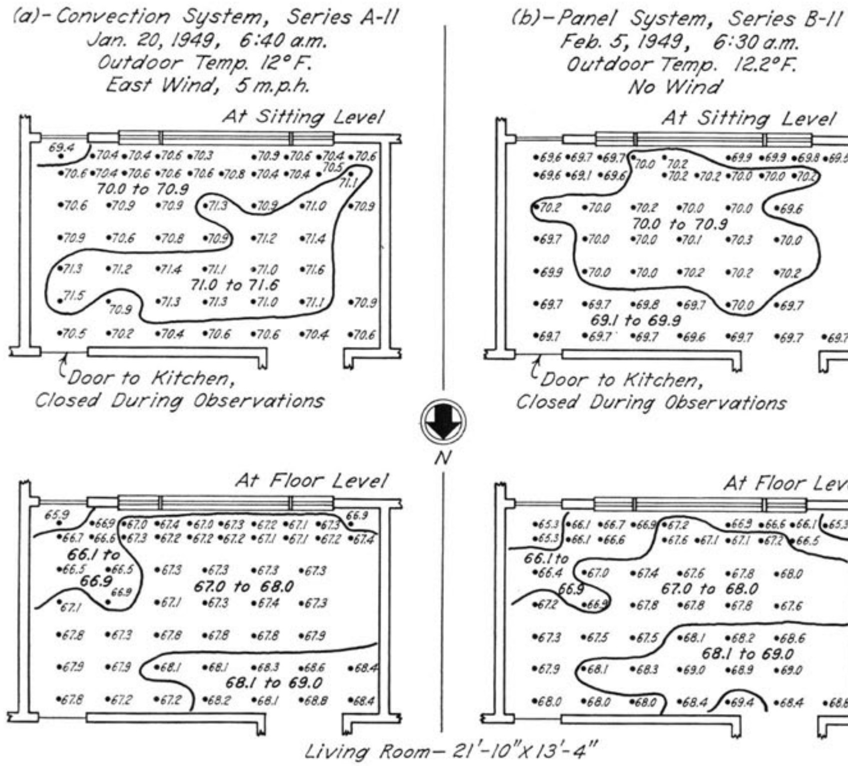


Fig. 17. Isothermal Contours in Living Room (Unheated Basement)

28. Room-Air Temperature Differentials

Room-air temperature gradients for each of the first-story rooms are shown in Fig. 18 for outdoor temperatures of about 35 F. The temperature differentials in the living zone of each room were slightly greater for the convection system than for the panel system, but a marked similarity in gradients existed for the two systems. As was noted in the previous year, although the thermostat setting was the same for the two systems, slightly lower room-air temperatures were maintained with the panel system.

The floor-surface temperatures for both systems were usually greater than the air temperatures at the floor level. Furthermore, a comparison of Fig. 18 for the unheated basement and Fig. 9 for the heated basement indicates that the floor-surface temperatures were 3 F to 4 F lower for the unheated basement. That this was true also for all the rooms of the Residence and over a wide range of weather conditions is indicated in Fig. 19, a summary of essential data on floor-surface temperatures.

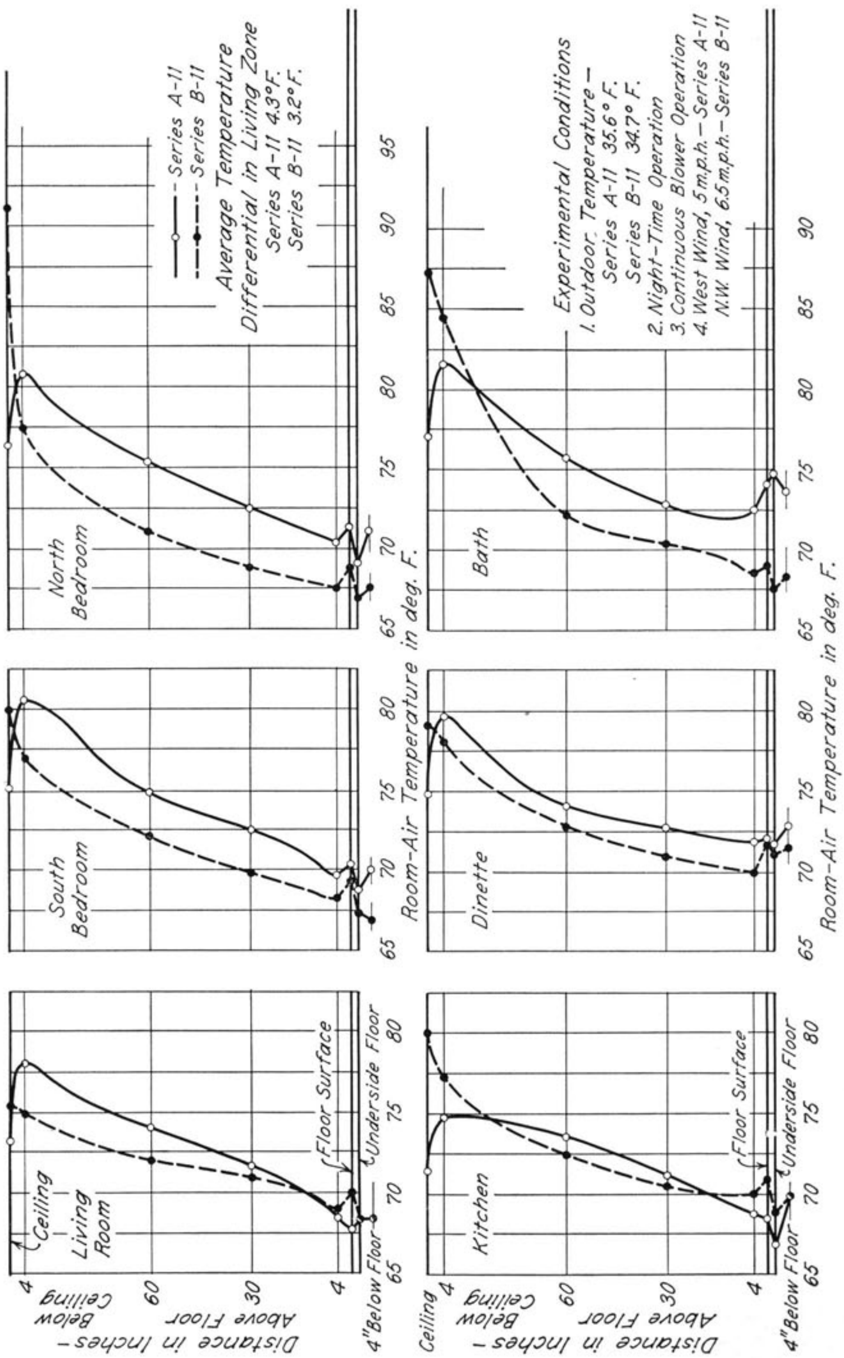


Fig. 18. Room-Air Temperature Gradients for an Outdoor Temperature of About 35 F (Unheated Basement)

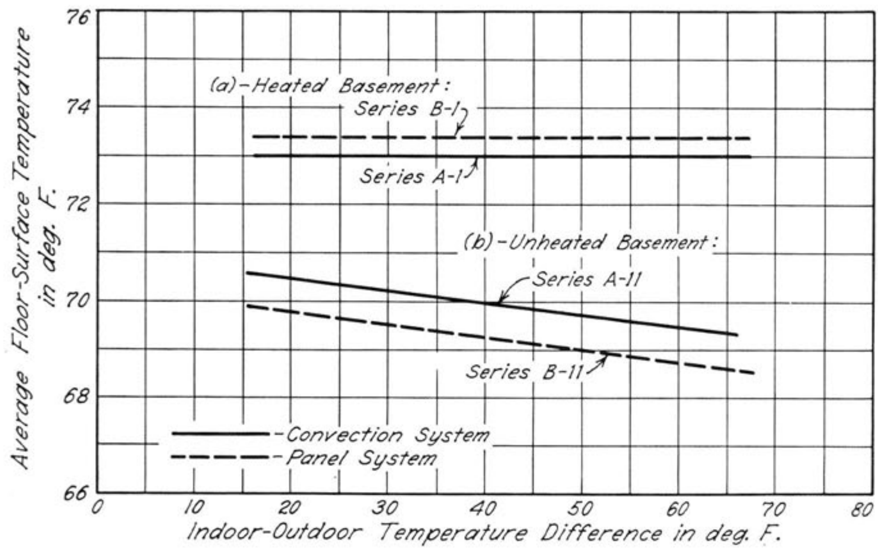


Fig. 19. Average Floor-Surface Temperatures

The average room-air temperature differentials for all rooms, shown in Fig. 20, indicate that the differentials for the panel system were slightly less than those for the convection system. A comparison of Fig. 20 with Fig. 11 indicates that the change from the heated to the unheated basement increased the room-air temperature differentials. For an indoor-outdoor temperature difference of 35 F the differential in the living zone increased from 2.5 F to 3.9 F for the convection system and from 1.5 F to 3.5 F for the panel system. The trends shown in Figs. 20 and 11 seem to imply that excessively large temperature differentials would be obtained when the subfloor space is maintained at temperatures much below 60 F, as in the case of crawl-space construction.

Also, in the case of the panel system the temperature differential between the breathing level and the ceiling level with an unheated basement was about twice that experienced with the heated basement. This large change could be accounted for by the higher ceiling-surface temperatures required with the unheated basement as compared to those for the heated basement (see Section 29).

From the preceding discussion it is apparent that the most significant effect of the change in basement-air temperature was to change the temperature differential from floor to breathing level as well as from floor to ceiling. Since the minimum differential was obtained when the basement was heated, the maintenance of a layer of warm air below the floor joists seems desirable. The heat loss from the furnace casing,

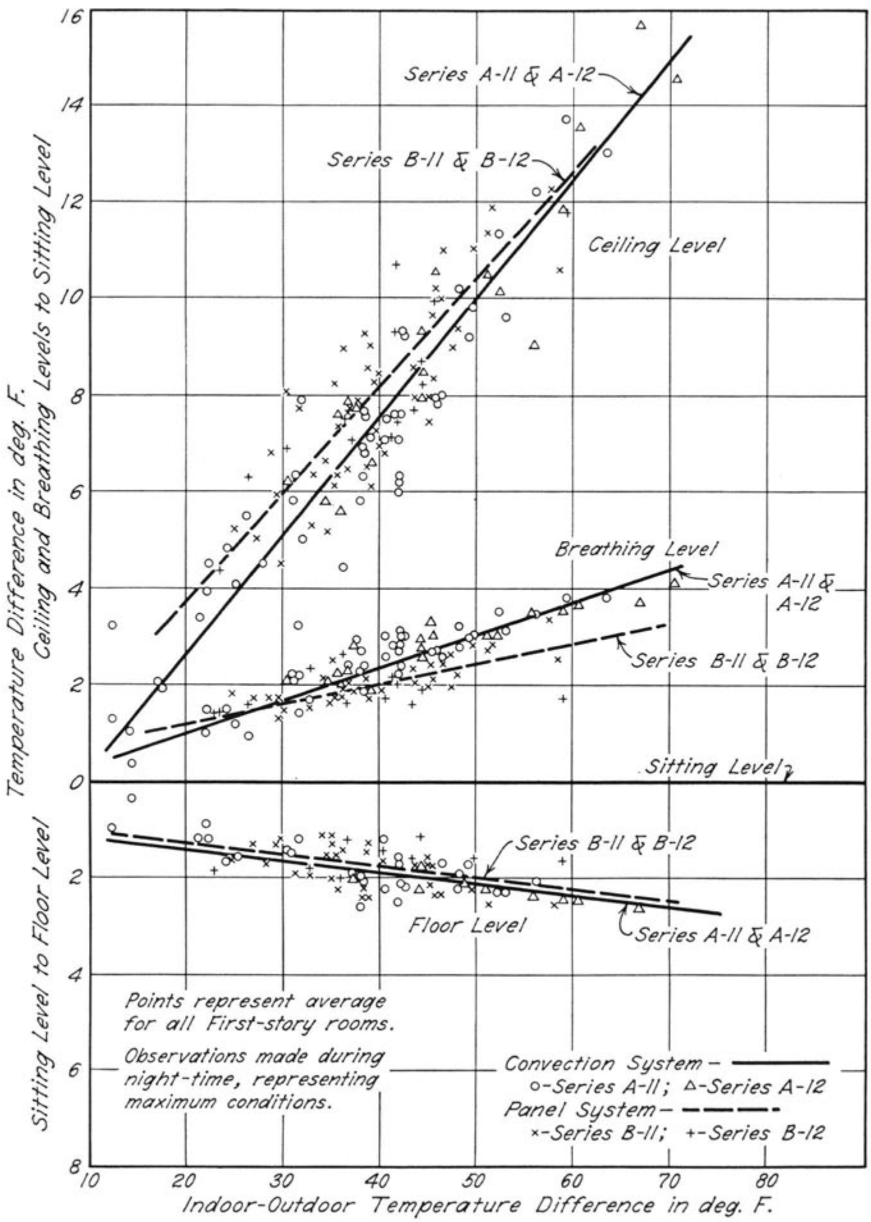


Fig. 20. Room-Air Temperature Differentials (Unheated Basement)

furnace bonnet, and ducts located in the basement cannot be regarded as true losses, since this heat regain provides a floor-panel effect and thereby affects the comfort in the rooms above the heating plant. In any case, whether by heat regain or by direct admission of heated air into the basement, it seems desirable to maintain relatively high temperatures of the air below the floor.

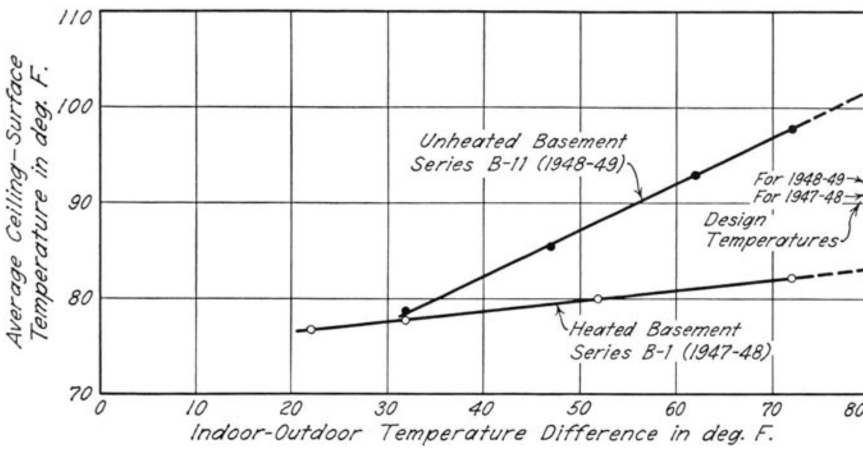


Fig. 21. Average Ceiling-Surface Temperatures for Panel System

29. Ceiling-Surface Temperatures Obtained with Panel System

The average panel-surface temperatures with both the heated basement (series B-1) and the unheated basement (series B-11) are shown in Fig. 21. The points designated by "Design Temperatures" indicate the calculated values of the panel-surface temperatures as listed in Table 2, columns 7 and 8.

A comparison of the curves with the corresponding design values indicates that with the heated basement the actual ceiling-surface temperature was lower than the design value, whereas the converse was true in the case of the unheated basement. The only rational explanation of this discrepancy seems to be in the magnitudes of heat regain in the two cases. Apparently, with the heated basement, the room heat losses were being satisfied from both ceiling-panel and floor-panel effects. Since the floor-panel effect was sizable in magnitude this would result in a corresponding decrease in ceiling-panel effect. On the other hand, in the case of the unheated basement, no floor-panel effect was observed. In fact, a reverse heat flow occurred through the floor which had to be compensated for by the ceiling panel alone. A further discussion of heat regains is given in Section 39.

30. Heat Transfer Through Floor

That a heat loss through the floor actually existed when the basement was unheated is shown more clearly by the temperatures indicated in Fig. 22. An exploratory survey of the floor-surface temperatures obtained with the unheated basement revealed three separate temperature areas.

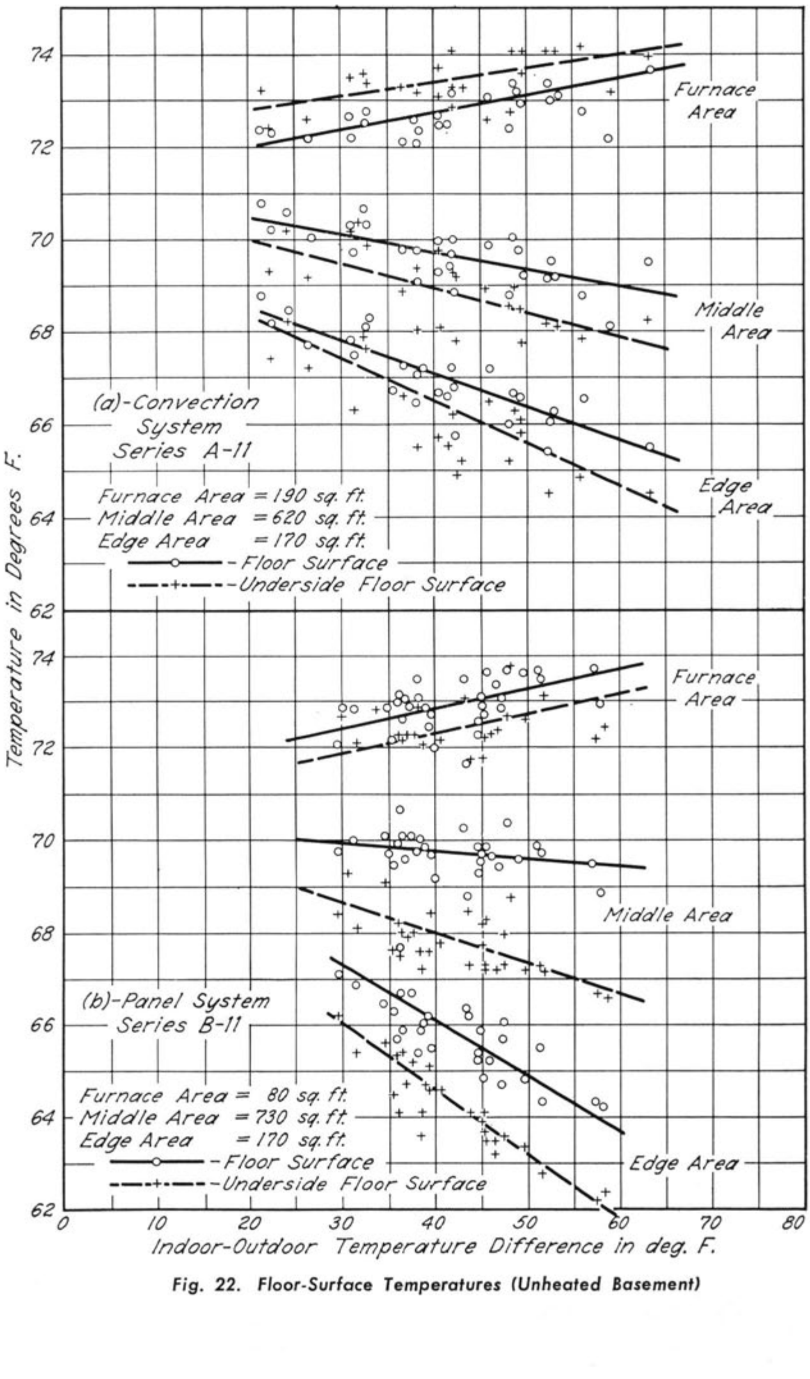


Fig. 22. Floor-Surface Temperatures (Unheated Basement)

The highest temperatures existed in the floor area above the furnace and the trunk ducts, or in the furnace area. The lowest temperatures prevailed in the edge area, which consisted of a 2-ft-wide section of the floor at the perimeter of the house. The rest — the middle area — constituted about 70 percent of the floor area.

Figure 22 shows the temperatures both for the floor surface (solid line) and for the underside of the floor surface (broken line). With the convection system the temperature difference between the upper and lower surfaces in the floor indicated that the heat transfer was upward into the living space in the furnace area and downward in the other areas. For the panel system, on the other hand, the heat transfer was downward in all areas and was of greater magnitude. With a heat transfer coefficient, or U value, for the floor of 0.89 Btu per hr (sq ft) (F), the average heat loss downward at an outdoor temperature of 35 F was only 410 Btu per hr for the convection system as compared to a loss of 1160 Btu per hr for the panel system. These results confirm the earlier statements that the heat regain from the furnace casing, etc., was larger for the convection system than for the panel system. The assumption that the floor loss may be neglected in designing a convection system in which no registers are provided for heating the basement was not entirely verified. However, the error involved in such a design may be considered negligible.

31. Mean Radiant Temperatures in Living Room

The importance of the surface temperatures on comfort was discussed in Section 15 in conjunction with an explanation of the distinction between average surface temperature, AST, and mean radiant temperature, MRT. Thus far in the discussion only the values of AST have been presented, primarily because of the ease in obtaining data. During the 1948-49 season, however, studies of the MRT for both the panel and convection systems were conducted by means of a thermo-integrator,[13] an instrument for determining the MRT of surrounding surfaces; measurements were made at only one location in the living room. The instrument was placed at the center of the room and hence the data were representative of the radiation effect experienced at that one location. Preliminary observations made with smoke currents showed that free-convection air flow existed at the surface of the instrument for both the panel and convection systems. Since the integrator could "see" not only the walls, floors, and ceiling but also the furniture, blinds, draperies, and other objects in the room, the MRT provides a better index of the radiation effect than does the AST.

The values of MRT determined with the thermo-integrator are shown in Table 6. For both systems the MRT differed only slightly from the ambient room-air temperatures. The room-air temperature was 0.3 F higher than the MRT for the convection system and was 0.5 F lower than the MRT for the panel system, indicating slightly different ratios of convection and radiation losses from the integrator. In any case the small total difference in MRT, of the order of 0.8 F, between the two systems verified the conclusions arrived at earlier with AST measurements and showed the remarkable similarity in results obtained with the two systems.

Table 6
Mean Radiant Temperatures in Living Room

Series	Outdoor Temp., F	Mean Radiant Temp., F	Ambient Air Temp., F	Difference, F	Average Difference, F
		Convection System			
A-14	15	72.0	72.3	−0.3	
A-14	27	71.6	71.8	−0.2	
A-14	32	72.1	72.4	−0.3	
A-14	40	72.0	72.4	−0.4	−0.3
		Panel System			
B-11	20	72.0	71.3	+0.7	
B-11	26	71.9	70.9	+1.0	
B-11	35	71.8	71.3	+0.5	
B-11	40	72.4	72.1	+0.3	+0.5

32. Summary of Room Temperatures

From the observations made with the unheated basement of temperatures of room air, ceiling surface, and floor surface as well as MRT, it was concluded that so far as these factors were concerned the performances of the convection and panel systems were similar. The panel system produced a wider zone of minimum air temperatures near the exposed walls in the living room than did the convection system, even though the range of temperatures from maximum to minimum was about the same. The room-air temperature differentials in the living zone were less for the panel system than for the convection system. The mean radiant temperatures determined with the thermo-integrator were slightly higher for the panel system, and differed only slightly from the room-air temperatures.

Although these temperature conditions were satisfactory with the unheated basement, improved conditions were obtained for both systems when the basement was heated, in that smaller room-air temperature differentials and warmer floor-surface temperatures were maintained.

VIII. PLANT PERFORMANCE

33. Performance of Burner and Furnace

Seasonal performance curves for the burner and furnace are shown in Fig. 23a for the convection system and in Fig. 23b for the panel system. In both figures the solid lines (series A-11 and B-11) represent continuous blower operation, and the broken lines (series A-12 and B-12) represent intermittent blower operation. The comparisons between the intermittent and continuous blower operations as well as between panel and convection systems were essentially in agreement with those discussed in Section 18 for the heated basement. That is, continuous blower operation showed less fuel consumption than did intermittent blower operation. Also, the fuel consumption for the panel system was approximately 25 percent higher than for the convection system, a difference which is of the same order of magnitude as the 20 percent that was obtained in the case of the heated basement.

In Fig. 23b the curve for the total operation of the gas valve indicates that the fuel input rate would not have been sufficient to heat the house in design weather (80-F indoor-outdoor temperature difference). However, as mentioned in Section 24, this limitation had been anticipated.

A comparison of the fuel consumption data for series A-11 (Fig. 23) with those of series A-1 (Fig. 15) indicates that the fuel consumption was about 10 percent higher for the heated basement than for the unheated basement. This percentage increase is not as large as the 16 percent difference in total calculated heat losses listed in Table 1, but was considered to be in fair agreement. Actually this 10 percent increase was required in order to raise the air temperature of the basement only about 7 F, since even in the case of the unheated basement the heat regains were sufficient to maintain a temperature above 57 F even in the coldest weather. This temperature gain of the basement air was proportionally small in terms of the fuel consumption. However, the increased comfort obtained in the first-story rooms as a result of warmer floor surfaces and smaller temperature differentials in the living zone, together with the fact that the basement space was made livable, shows that the comfort gains were quite large.

34. Performance of Blower

Seasonal performance curves for the blower are shown in Fig. 24a for the convection system and Fig. 24b for the panel system. Essentially the performances were similar in all cases to those discussed in Section 19. The only noticeable difference was in the electrical input to the blower motor, which was substantially less with the unheated basement. This can be accounted for by the smaller air-flow rate used when the basement was unheated.

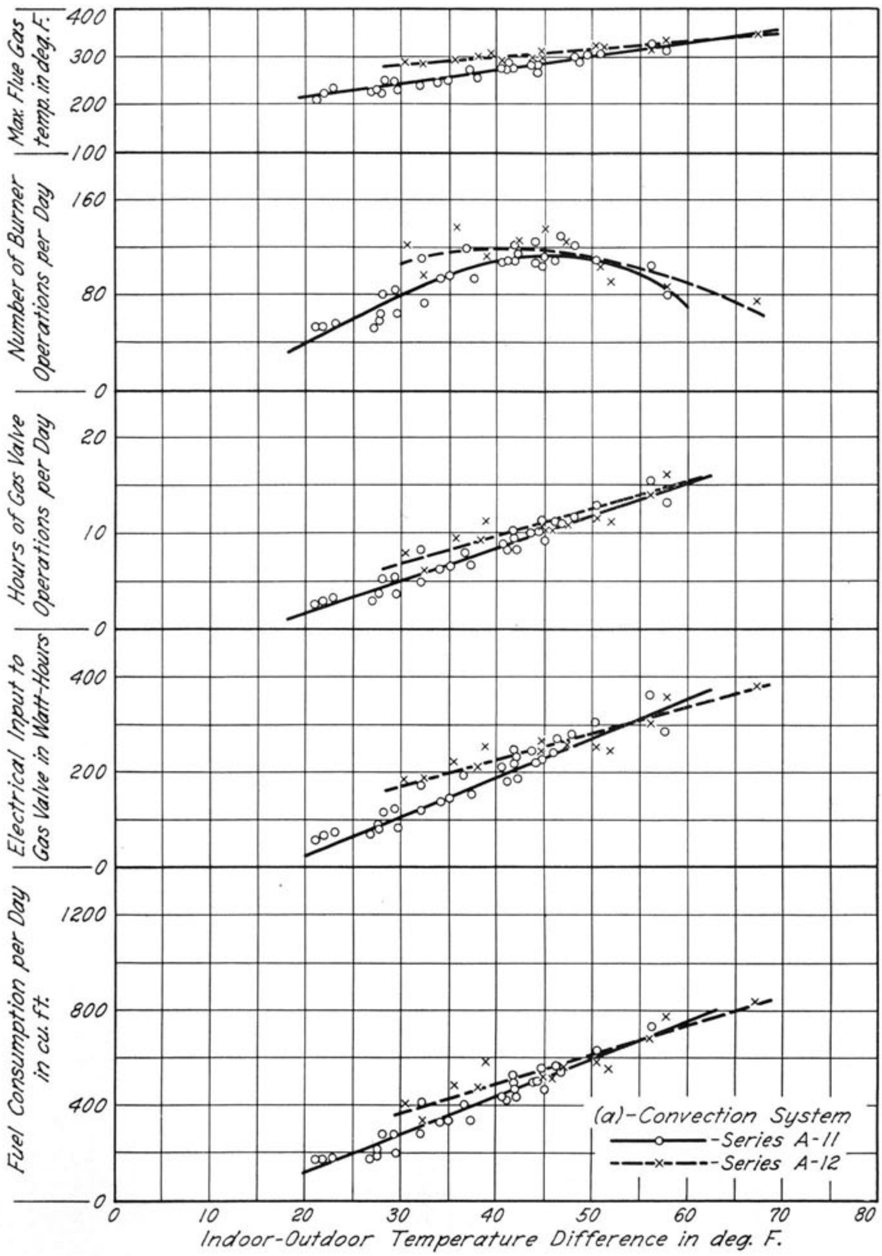


Fig. 23a. Performance of Burner and Furnace with Convection System (Unheated Basement)

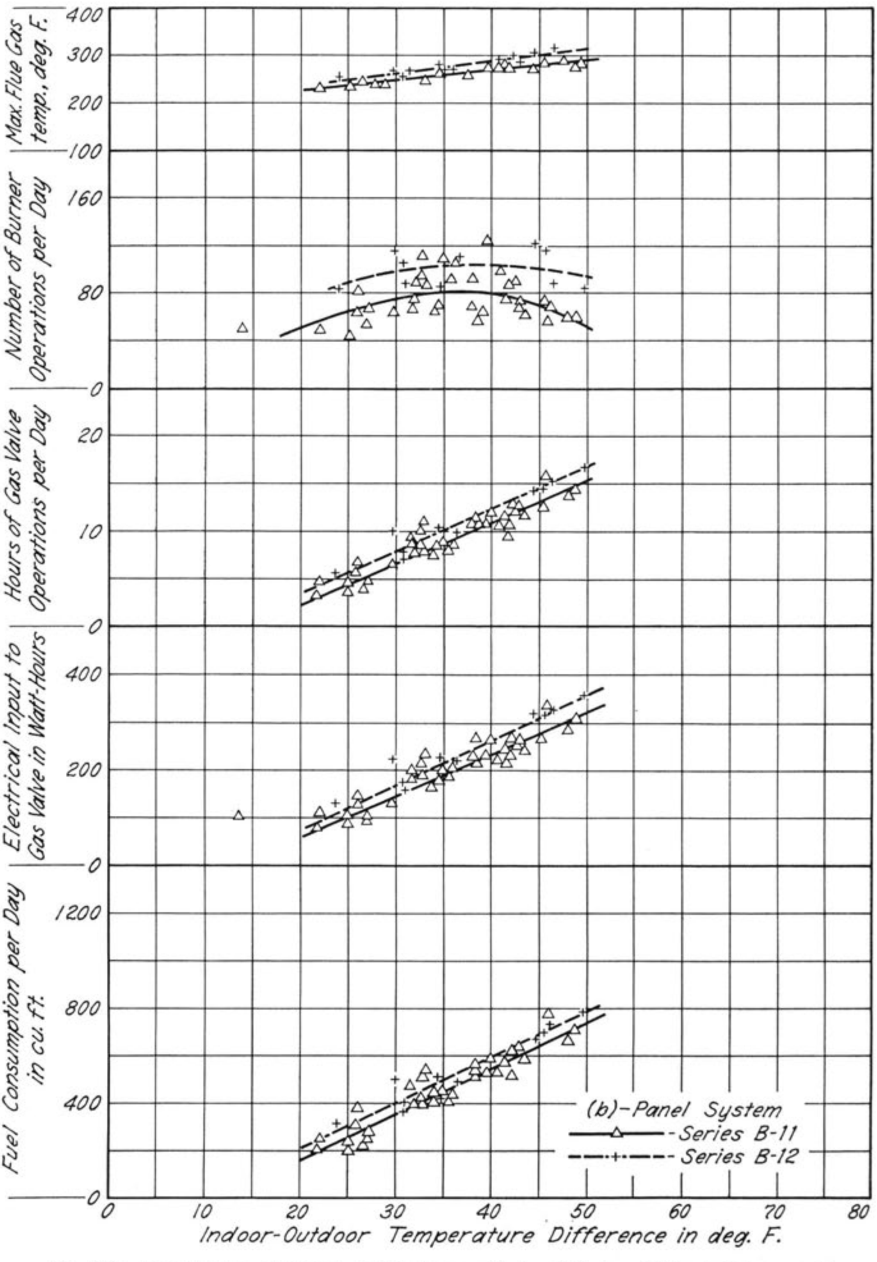


Fig. 23b. Performance of Burner and Furnace with Panel System (Unheated Basement)

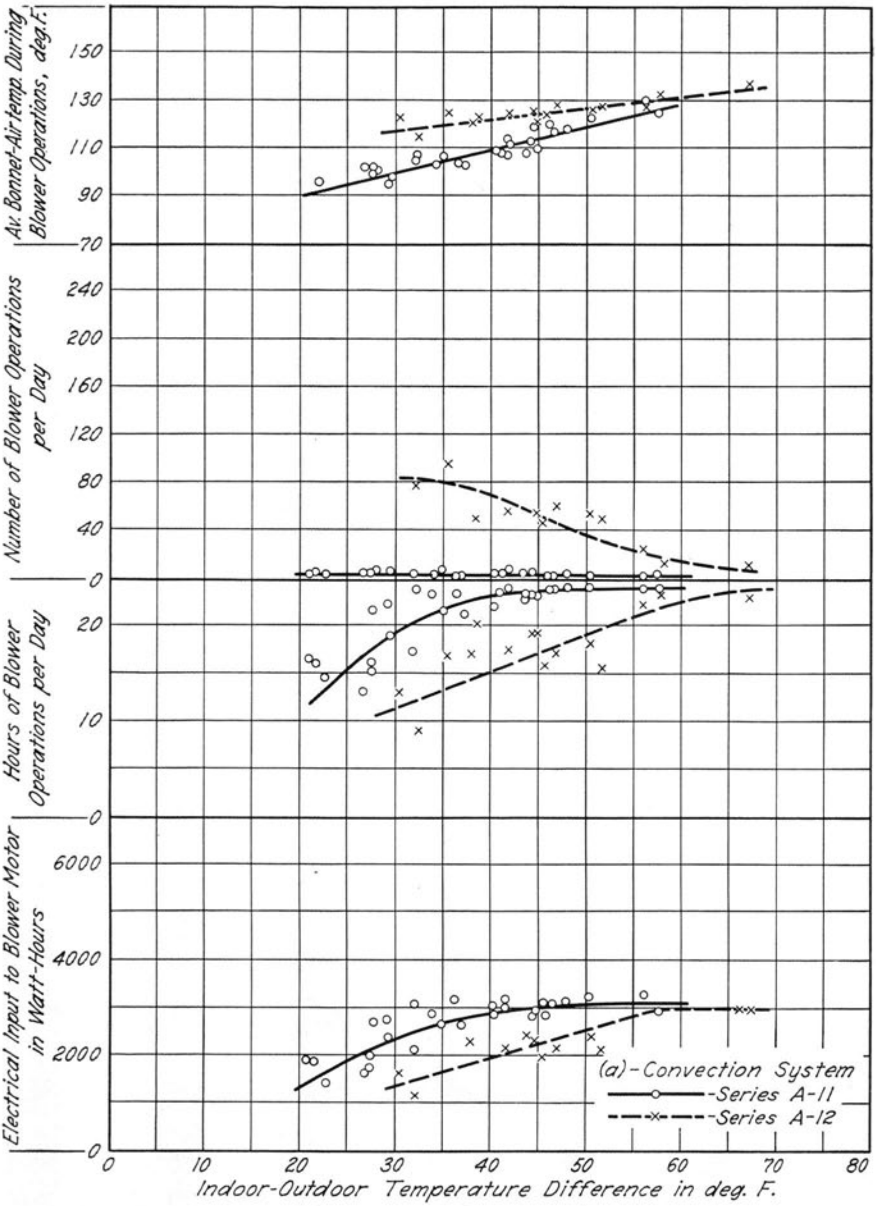


Fig. 24a. Performance of Blower with Convection System (Unheated Basement)

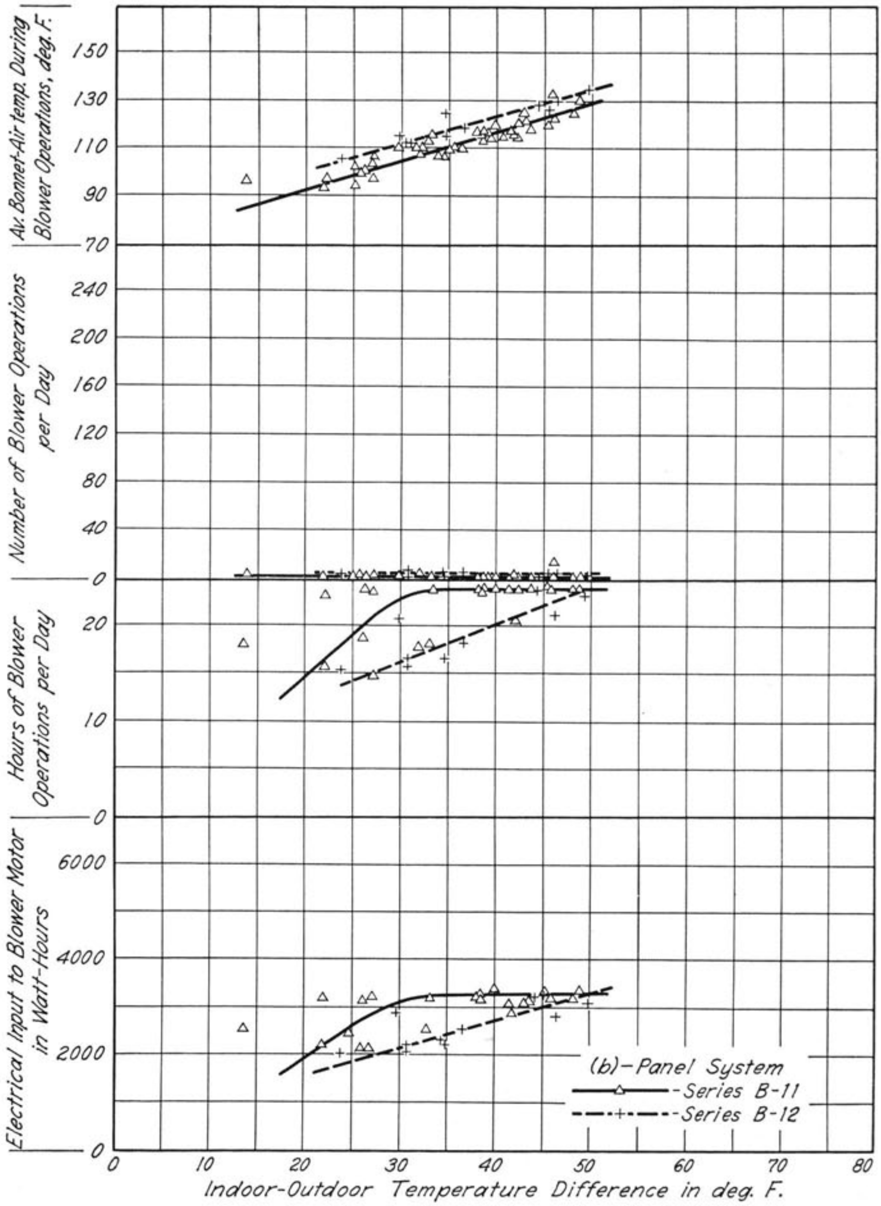


Fig. 24b. Performance of Blower with Panel System (Unheated Basement)

IX. PERFORMANCE WITH MAXIMUM DIFFERENTIAL SETTING OF THERMOSTAT

35. Room-Air Temperature Variations During Cycling of Burner and Blower

Following the continuous air circulation principle advocated by the National Warm Air Heating and Air Conditioning Association the importance of using a small temperature differential setting for the room thermostat is emphasized. To illustrate the differences in over-all plant performance with two widely different settings of the thermostat differential, two additional studies were conducted. Series A-13 was conducted with low settings of the fan switch and series A-14 with high settings of the fan switch. In both series the differential setting of the room thermostat was adjusted to provide the maximum temperature differential. These series are directly comparable with series A-11, for which the minimum differential setting was used.

Excerpts from the time-temperature charts are shown in Fig. 25. The sensitivity of the resistance thermometers used for recording the room-air temperatures at the sitting level is illustrated by the chart scale, which was $\frac{9}{16}$ in. per F change in temperature. Hence any slight variation in room-air temperature was greatly magnified. Series A-11, utilizing a minimum differential setting, was used as a basis of comparison. As may be noted from the top chart in Fig. 25, practically no variation in temperature was observed with series A-11. The middle chart shows the changes caused by increasing the differential setting of the thermostat to its maximum value. The length of burner operation was increased from about 4 min to about 32 min; the frequency of operation was reduced from about 6 per hr to about 1 per hr. The resulting maximum variation in room-air temperature was of the order of 0.8 F.

The bottom chart in Fig. 25 shows the combined effect of the maximum setting of the thermostat differential and the high settings of the fan switch. In this case the maximum variation in room-air temperature was 1.2 F. It is probable that the temperature variations shown in Fig. 25 were smaller than would normally be experienced in a house that is not as well weatherproofed as the Residence.

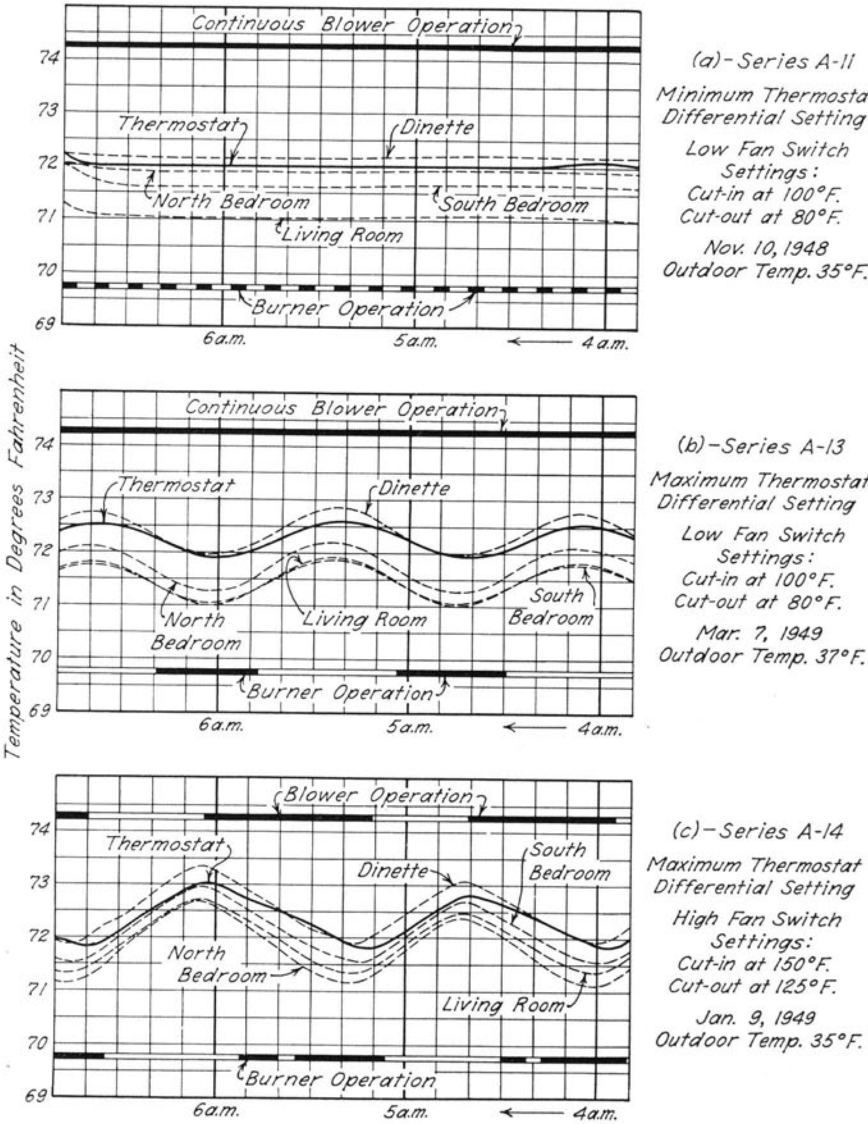


Fig. 25. Room-Air Temperature Variations with Two Settings of Room Thermostat Differential

36. Room-Air Temperature Differentials

In the case of series A-11, for which the minimum thermostat differential was used, the temperature differentials in the rooms remained constant with respect to time for any given weather condition. That is, the difference in temperature between the sitting level and floor level, for example, was the same at both the beginning and the end of the burner cycle. Hence, as shown in Fig. 20, a single line could be used to represent the air temperatures at the floor level over the range of outdoor temperatures experienced. However, in the case of series A-13 and A-14, for which the maximum thermostat differential was used, the temperatures at any given level were not constant with respect to time. In fact, the difference in temperature between the sitting level and the floor level was larger at the end of the burner operation than at the beginning. Therefore, in plotting the room-air temperature differentials in Fig. 26, a large scattering of the observed points was inevitable. The three pairs of broken lines in Fig. 26 indicate the range of this temperature variation at the ceiling, breathing, and floor levels. As far as the average room-air temperature differentials from the sitting level were concerned, no appreciable change was noted as compared with series A-11.

The evidence shown in Figs. 25 and 26 indicates that in order to maintain a uniform room-air temperature it was necessary to use the minimum setting of the thermostat differential. To a lesser extent a low setting of the fan switch was also found to be desirable.

37. Performance of Burner and Furnace, and Blower

Complete performance curves similar to Figs. 23 and 24 for series A-11 were also plotted for series A-13 and A-14, but since the results were practically the same they are not included in this bulletin. A slight increase in fuel consumption was indicated when the room thermostat differential was increased, but the increase was within the range of experimental deviations. The only significant change was in the number of burner and blower operations, which as indicated in Fig. 25 were relatively infrequent when the room thermostat differential was increased.

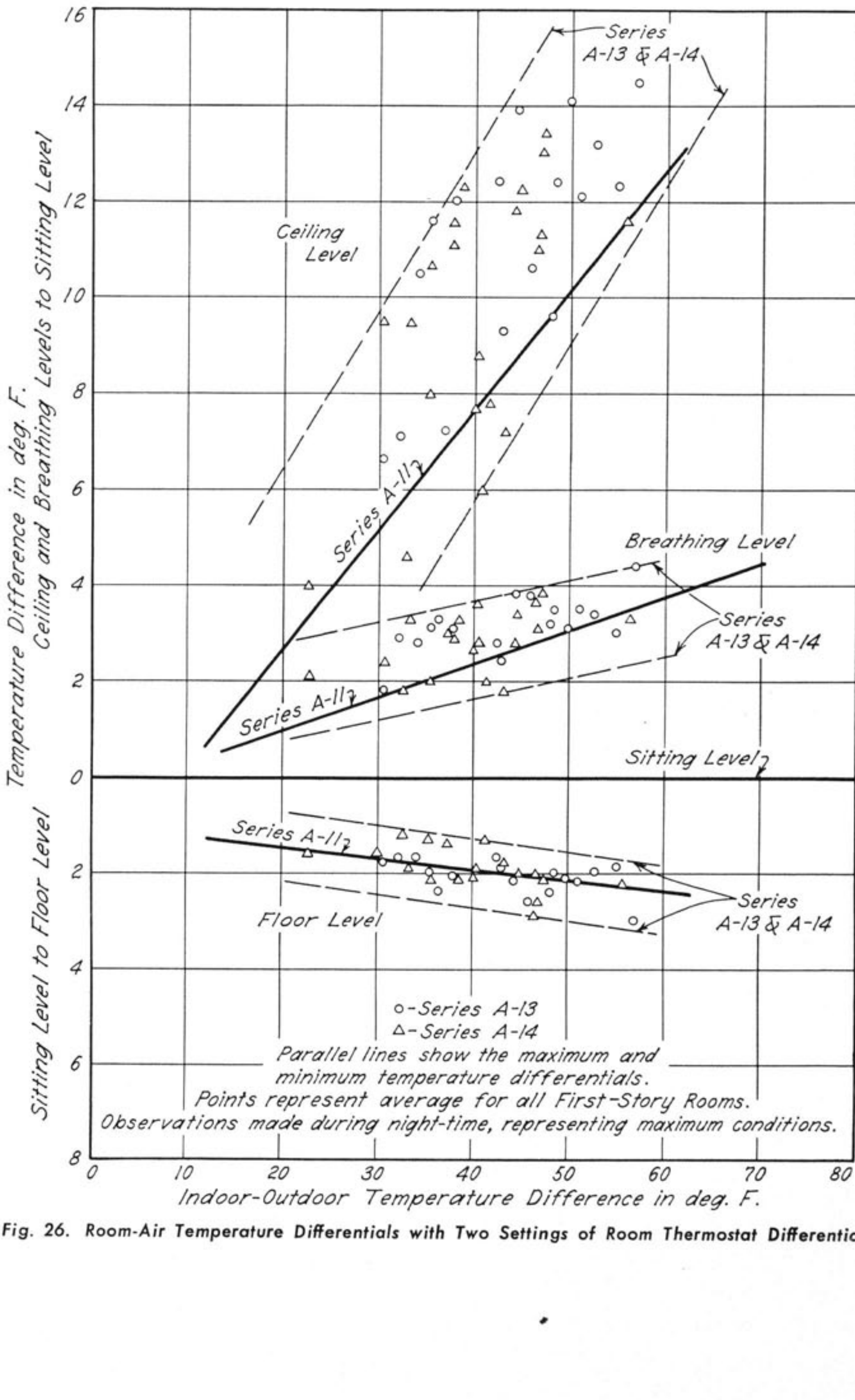


Fig. 26. Room-Air Temperature Differentials with Two Settings of Room Thermostat Differential

GENERAL DISCUSSION

X. HEAT INPUT RATES AND HEAT REGAINS

38. Total Heat Input Rates and Calculated Losses

The selection of the proper size of furnace and the adjustment of the fuel input rate are of importance to the installer because an oversized furnace may be initially costly and its use may result in inefficient operation. Furnace sizing is of particular concern to the gas industry, since the increments in capacity rating of successive sizes of furnaces are smaller than those commonly employed in the cases of both oil-fired and solid-fuel burning furnaces. Although the primary objective of this investigation did not include a study of the factors affecting the selection of furnace sizes or fuel input rates, nevertheless the experience obtained with the unheated basement (series A-11) indicated that an analysis of the calculated heat losses of the Residence and the total heat input rates should be made. In this case, as indicated later (page 81), the fuel input rate proved to be insufficient for design weather conditions.

The experience in Research Residence No. 1 had shown that the numerous assumptions made in the design procedure for a heating system were not always verified in the actual operation of the system. That is, the distribution of the heat liberated in the furnace to the various parts of the Residence was markedly affected by the magnitude and source of the heat regains from furnace casing, furnace bonnet, flue pipe, and ducts. The net result was that the values assumed in the design were frequently found to deviate considerably from actual measured values. For Research Residence No. 2 it was considered desirable, therefore, to make an analysis (as far as available data would permit) of total heat input rates, calculated heat losses, and deviations from design assumptions.

A complete tabulation of items referring to calculated heat losses, heat input rates, bonnet capacities, air-flow rates, and temperature rises is shown in Table 7 for the heated basement and in Table 8 for the unheated basement. A graphical representation of the essential heat transfer rates with both the heated and unheated basement is shown in Fig. 27. All values given in this chapter and in Fig. 27 apply specifically

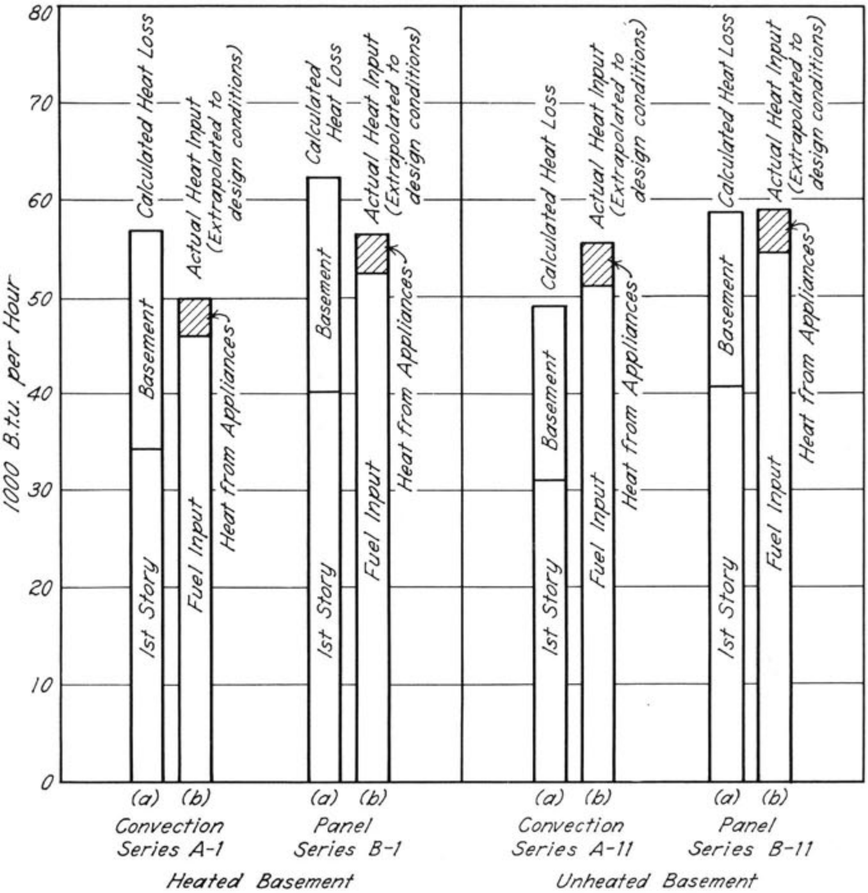


Fig. 27. Comparison of Heat Losses and Heat Inputs

to design weather conditions. The calculated heat losses for series A-1, B-1, A-11, and B-11 are designated by bar graphs (a). The total calculated heat losses shown in Fig. 27 correspond to those listed under item 4 (b) in Tables 7 and 8 and represent heat losses as usually calculated for design weather conditions. The difference between the calculated values of the first-story heat losses for series A-1 and A-11 has been discussed in Section 23; it does not represent changes in actual heat loss.

In any comparative study of actual heat input rates and calculated heat losses the only defensible approach is that in which the entire structure, including the basement, is considered in the heat loss calculations. It is true that in the 1948-49 investigation no heated air was

Table 7

Relation Between Design and Actual Operating Conditions for 1947-48 Investigation

	Convection System			Panel System		
	Design Conditions	Actual Continuous Burner and Blower Operation	Actual Operation of Burner and Continuous Operation of Blower	Design Conditions	Actual Continuous Burner and Blower Operation	Actual Operation of Burner and Continuous Operation of Blower
	(1)	(2)	(3)	(4)	(5)	(6)
1) Design Temperature Difference, F	80	......	80	80	......	80
2) Calculated Heat Loss, Btu per hr (First Story)	34 497	......	......	40 102	......	......
3) Calculated Heat Loss, Btu per hr (Basement)	26 285	......	......	26 285	......	......
4) Total Calculated Loss, Btu per hr a) With 20% Reserve	60 782	......	......	66 387	......	......
b) With No Reserve	56 930	......	......	62 540	......	......
5) Bonnet Efficiency, percent	80	......	......	80	......	......
6) Duct Transmission Efficiency, percent	100	......	......	85	......	......
7) Fuel Input Rate, Btu per hr	76 000	76 000	50 000 required	97 600	76 000	59 600 required
8) Rate of Heat Input from Appliances, Btu per hr	......	......	3 700	......	......	4 040
9) Total Heat Input Rate, Btu per hr	......	......	53 700	......	......	63 640
10) Bonnet Capacity, Btu per hr	60 782	60 800	38 000	......	60 800	42 600
11) Weight Rate of Air Flow, lb per hr	2 500	2 500	2 500	3 580	3 580	3 580
12) Air Flow Rate, cfm	565	565	565	795	795	795
13) Temperature Rise, F	100	100	62	70	70	49

Columns 2 and 5—Values shown are for actual steady-state conditions, in which both the burner and blower were operated continuously.

Columns 3 and 6—Values shown are based on actual data obtained under normal operating conditions; the observed values for 24-hr periods were extrapolated to design weather conditions.

Item 2, column 4—Includes loss upward to attic and outward through edges of panel.

Item 3—This includes 3600 Btu per hr for a reserve capacity in basement, in accordance with the calculation procedure given in *Manuel 3* of the National Warm Air Heating and Air Conditioning Association. However, in actual operation, the basement temperature was only about 68 F, so that reserve capacity was not required.

Item 4(a)—This is a summation of items 2 and 3 and includes the 20 percent reserve for basement rooms and bath.

Item 4(b)—This item is the result of subtracting the 20 percent reserve allowed for basement rooms and bath from item 4(a). This represents a closer approximation than does item 4(a) of the calculated heat loss for the entire structure and will be compared with actual heat inputs.

Item 7, columns 2 and 5—Fuel inputs for panel system were set at the same value as for the convection system as discussed in Section 7. The fuel input was based upon item 4(a), column 1.

Item 8—This includes heat from lighting, cooking, and electrical appliances. It does not include solar gain or heat from occupants.

Item 9—This is a summation of items 7 and 8.

Item 12—Based upon an air density of 0.075 lb per cu ft.

Table 8

Relation Between Design and Actual Operating Conditions for 1948-49 Investigation

	Convection System			Panel System		
	Design Conditions	Actual Continuous Burner and Blower Operation	Actual Operation of Burner and Continuous Operation of Blower	Design Conditions	Actual Continuous Burner and Blower Operation	Actual Operation of Burner and Continuous Operation of Blower
	(1)	(2)	(3)	(4)	(5)	(6)
1) Design Temperature Difference, F	80	......	80	80	......	80
2) Calculated Heat Loss, Btu per hr (First Story)	31 047	......	......	40 625	......	......
3) Calculated Heat Loss, Btu per hr (Basement)	21 300	......	......	21 300	......	......
4) Total Calculated Loss, Btu per hr a) With 20% Reserve	52 347	......	......	61 925	......	......
b) With No Reserve	49 380	......	......	58 950	......	......
5) Bonnet Efficiency, percent	80	......	......	80	......	......
6) Duct Transmission Efficiency, percent	85	......	......	85	......	......
7) Fuel Input Rate, Btu per hr	46 000	46 000	46 700 required	59 700	46 000	58 400 required
8) Rate of Heat Input from Appliances, Btu per hr	......	......	4 530	......	......	4 710
9) Total Heat Input Rate, Btu per hr	......	......	51 230	......	......	63 110
10) Bonnet Capacity, Btu per hr	36 800	36 800	36 800	......	37 100	37 100
11) Weight Rate of Air Flow, lb per hr	1 530	1 530	1 530	2 180	2 180	2 180
12) Air Flow Rate, cfm	340	340	340	485	485	485
13) Temperature Rise, F	100	100	Over 100 required	70	70	Over 70 required

Columns 2 and 5—Values shown are for actual steady-state conditions, in which both burner and blower were operated continuously.

Columns 3 and 6—Values shown are based upon actual data obtained under normal operating conditions; the observed values for 24-hr periods were extrapolated to design weather conditions.

Item 2, column 4—Includes loss upward to attic and outward through edges of panel. This also includes the floor loss.

Item 3—This includes 2720 Btu per hr for a reserve capacity in basement, in accordance with the calculation procedure given in *Manual 3* of the National Warm Air Heating and Air Conditioning Association. However, in actual operation, the basement temperature was only about 60 F, so that reserve capacity was not required.

Item 4(a)—This is a summation of items 2 and 3 and includes the 20 percent reserve for basement rooms and bath.

Item 4(b)—This item is the result of subtracting the 20 percent reserve allowed for basement rooms and bath from item 4(a). This represents a closer approximation than does item 4(a) of the calculated heat loss for the entire structure and will be compared with actual heat inputs.

Item 7, columns 2 and 5—Fuel inputs for panel system were set at the same value as for the convection system as discussed in Section 23. Fuel input was based upon item 2, column 1.

Item 8—This includes heat from lighting, cooking, and electrical appliances. It does not include solar gain or heat from occupants.

Item 9—This is a summation of items 7 and 8.

Item 12—Based upon an air density of 0.075 lb per cu ft.

Item 13, columns 3 and 6—As discussed in the text (page 81), the actual heat input of 46,000 Btu per hr was not quite sufficient to meet the demand. By extrapolation of data, it would appear that a slightly higher temperature rise would have been necessary on a design day for the air-flow rate established for the systems. This would have required the slightly higher fuel input rates shown in item 7, columns 3 and 6.

introduced into the basement through registers. However, as mentioned previously (Section 24), the basement-air temperature did not drop below 57 F, even in the coldest weather. Hence, strictly speaking, the basement was not unheated. In addition, a sizable heat loss occurred from the basement to the outdoors, and therefore a substantial portion of the total heat input rate was dissipated in this manner. This calculated basement loss varied from about 40 percent of the total heat loss for series A-1 to 32 percent for series B-11. It is possible that these percentages are higher than those obtained in houses having more conventional window construction in the basement, even though the full-size windows used in the Residence were weatherstripped and equipped with storm sash.

The bar graphs (b) in Fig. 27 represent the total heat input rates to the Residence. They consist of the fuel input rate and the heat input rate from lighting, cooking, and electrical appliances but do not include solar heat gain or heat gain from occupants, nor do they take into account the heat loss from the flue gas at the attic floor level of the chimney. The values for the fuel input rate (item 7, columns 3 and 6) in Tables 7 and 8 would have been required for design weather conditions and were based upon extrapolation of the fuel consumption curves given in Figs. 15 and 24. These are maximum values that represent the greatest deviations from the average curves shown in those figures.

The ratios of the calculated heat loss to the total heat input rate were 1.07 for the convection system and 0.98 for the panel system; both were for the heated basement, indicating that the calculated losses were slightly higher than the total heat input rates. If in the 1947-48 calculations of heat loss the infiltration loss through doors had been assumed to be the same as that used in the 1948-49 calculations, these same ratios would have been 1.00 and 0.93 for the heated basement. Similar ratios for the unheated basement were 0.96 for the convection system and 0.93 for the panel system. It is realized that the heat loss calculations involve a number of assumptions, so that the magnitudes of heat losses can only be regarded as approximations. Nevertheless, it is significant that the calculated total heat losses for the entire structure were of the same order of magnitude as the total heat input rates.

In earlier days of the central heating of homes the problem of selecting a furnace having adequate capacity was relatively simple because the furnace was always located in the basement and a predominant majority of installations used solid-fuel burning equipment for which the operating capacity was not fixed at a single value. In modern practice, however, in which gas-fired equipment has been used to a greater extent than previously, the increments of fuel input rates for each increase in size of furnaces produced by a given manufacturer may be as small as 20,000 Btu per hr. Furthermore, it is possible to adjust the fuel input rate for any given furnace at the pressure regulator or main gas valve. The net result is that the fuel input rates can be selected as close to the predicted design requirements as the heating contractor may choose.

An even greater change in heating practice during recent years has resulted from the great diversity of building practice and furnace locations. To cite a few cases that are encountered in practice, the following deviations can be given.

(a) Houses built without a basement and provided with a crawl space.

(b) Houses built without a basement and upon a concrete floor slab laid on the ground.

(c) Furnaces installed in an enclosed closet, vented to the attic or outdoors.

(d) Furnaces installed in a crawl space that is vented to the outdoors.

(e) Furnaces installed in an attic that is ventilated.

(f) Furnaces installed in a separate heater room in the garage.

(g) Heating systems provided with outdoor-air ventilation ducts.

(h) Furnaces installed in homes having both heated and unheated basements.

Suitable equations for selecting the maximum fuel input rate and the proper size of furnace could be derived for each of the many variations listed. To do so, however, would result in eventual confusion and the use of a number of empirical rules that would apply only to limited cases. It is realized that, from practical considerations, a simpler method is preferable to a more detailed procedure. If, however, a single general equation can be adopted that would involve the fewest practical assumptions, such an equation would be preferable to a series of equations and factors for each special case, no matter how simple each equation may be.

The most logical general equation for use in selecting the maximum fuel input rate and furnace size for any condition of house construction or furnace location would be that based on treating the entire house and the furnace as an integral unit. In other words, the entire house can be considered as a calorimeter in which the furnace is the fuel-burning device.

The general equation that will apply to any residential warm-air heating system is

$$H_i = \frac{H_a + H_b + H_v}{(e_b)\ (e_t)} \tag{1}$$

in which

H_i = fuel input rate required for design weather conditions, Btu per hr.

H_a = design heat loss for spaces above first-story floor level, including normal infiltration losses, Btu per hr.

H_b = design heat loss for spaces and construction below the first-story floor level, Btu per hr. In the case of slab-floor construction the heat loss through the floor would be included. If a basement space is heated to 70 F the normal infiltration heat loss for the basement should be included. If the basement is not heated to 70 F an approximation should be made of the expected basement-air temperature. In the Residence studies a value of 60 F was assumed.

H_v = ventilation heat loss, Btu per hr. Studies in progress at the time of writing this bulletin indicate that where outdoor air for ventilation is used the ventilation heat loss should be included and that no reduction should be made in the normal infiltration heat losses calculated in items H_a and H_b.

e_b = bonnet efficiency. For approved gas-fired forced-air furnaces* the rated bonnet efficiency is 80 percent. For an actual installation, however, in which the heat losses from the furnace casing, furnace bonnet, and flue pipe are completely regained in the spaces included in the calculations for H_a and H_b, a value of 90 percent is suggested in place of the rated value of 80 percent. For those installations in which the heat regain will be small—such as for closet-furnace installations vented to the attic and for furnaces located in the attic, crawl space, or outside the structure — the rated value of 80 percent is suggested for use.

e_t = duct transmission efficiency. If the trunk duct and branch ducts are located in the spaces included in the calculations for H_a and H_b, a value of 100 percent is suggested, since the heat loss from ductwork and stacks is regained. For duct systems located in attic spaces or vented crawl spaces a value of 90 percent is suggested for insulated ducts and a lower value for uninsulated ducts.

It is true that the evaluation of item H_b is not precise: numerous assumptions are necessary in calculating basement heat losses, particularly those for walls and floors below grade. Nevertheless, the results obtained in the Residence indicate that the common practice of ignoring

* See Section 4, page 11 — definition of bonnet efficiency.

the basement heat loss for an unheated basement may involve difficulties due to insufficient input rates. For example, in the case of the unheated basement (Section 24) the fuel input rate for the convection system was determined by dividing the heat loss of the first-story rooms only, by the assumed values of bonnet efficiency and duct transmission efficiency. In this case, although the predicted register delivery was made equal to the calculated heat loss of the first-story rooms alone, the fuel input rate (46,000 Btu per hr) was less than both the total calculated heat losses for the entire Residence (52,347 Btu per hr) and the total heat input rates (51,230 Btu per hr). In other words, the fuel input rate was not quite sufficient for the heat loss requirements of the entire structure.

The value of 90 percent for e_b was selected as most nearly representative of the results obtained in the Residence studies. The application of the suggested equation to the four main series of studies conducted in the 1947-48 and 1948-49 heating seasons indicated that the fuel input rates should have been of the order of 67,500 Btu per hr for 1947-48 and 58,200 Btu per hr for 1948-49. The values actually used were 76,000 Btu per hr and 46,000 Btu per hr. The fuel input rates based on the suggested equation would have more nearly corresponded to the actual total heat input rates and still provided a reserve capacity.

39. Heat Regain as It Affects Design Considerations

Reference has been made at various points in this bulletin to the direct and indirect heat regain and its possible effect on the over-all performance of the heating system. Previous experience with two other Research Residences has shown that a substantial part of the heat required to offset the heat loss from the structure is from *indirect* sources such as the furnace casing, furnace bonnet, basement ducts, wall stacks, flue pipe, and chimney. In addition, heat gains occur from *direct* sources such as lights, cooking, appliances, sun, and occupants. It is true that in the case of a furnace located in the basement the heat gain from the furnace casing, furnace bonnet, and basement ducts will not be directly reflected as a heat gain to first-story rooms. However, any such heat gain to the basement, together with any direct addition of heated air through registers — which will result in a layer of heated air under any area of the subfloor — will produce a floor-panel heating effect in that portion of the first-story rooms. This effect was experienced with the convection system when the basement was heated as discussed in Sections 14 and 15. To a lesser extent the same effect was observed with the convection system when the basement was unheated, as discussed in Section 30. In the case of the ceiling panel system a floor-panel heating effect was observed with the heated basement but not with the unheated basement. In a similar manner, any portion of the wall surface which is heated by wall stacks or the chimney will result in a wall-panel effect. Strictly speaking, the convection system can be considered not as a true convection system but as a combined convection-panel system in which some amount of radiation heat transfer occurs. Furthermore, the panel system can be considered not as a true radiation system but as a combined radiation-convection system. This may serve as a partial explanation of the observed fact that the factors affecting comfort were not appreciably different for the two systems.

The heat gains from direct and indirect sources may also explain the discrepancies observed between the design values of plant performance and actual observed values. In the design of the heating system such direct and indirect sources of heat gain are ignored. In practice, however, the heat gain from these sources will supplement that entering through the registers or panels to offset the heat loss of the structure and thus will satisfy the thermostat before the system is required to operate under the design condition. The net result will be that the actual fuel input rate will be less than calculations had indicated would be required, as shown in Table 7, item 7. In addition the temperature of the circulating air will be substantially lower than design values, as shown in Table 7, item 13. It was not possible to make similar comparisons from the data shown in Table 8, since (see Section 38) the fuel input rate was not sufficient to meet the design weather demand.

In view of the large magnitude of the direct and indirect heat gains, some question might arise as to why they are not taken into account in the design of heating systems. Theoretically this can be done by calculating the heat loss by the usual method, subtracting the anticipated direct and indirect heat gains, and using the resulting value as the heat to be supplied to the structure through registers or panel surfaces. In practice, however, this introduces complications which do not warrant the necessary time and effort. For example, these heat gains cannot be predicted exactly on the basis of existing data. Even if such predictions could be made, it would be unwise to base the selection of equipment and fuel input rates on the minimum requirements. This is particularly true because the magnitude of the direct gains is not constant, being somewhat lower at night, on cloudy days, and on days when there is little cooking. Therefore both the direct and the indirect gains are normally considered as a factor of safety and are ignored in design. The fact remains, however, that whether or not these gains are taken into account in the design, they do affect the actual operation of the plant. In the case of the Residence, these gains not only caused the temperature of the circulating air for both systems to be lower than design values, but also caused the difference between the performances of the two systems to be less than was originally anticipated.

XI. SUMMARY AND CONCLUSIONS

This bulletin presents the first results obtained in Warm-Air Heating Research Residence No. 2, which was a one-story structure of frame construction with a large amount of glass exposure and with a full basement. The convection system that served as a basis for comparison was a conventional forced warm-air heating system which included an extended-plenum trunk duct and high-sidewall registers. The panel system was of the warm-air ceiling panel type in which the depth of panel space, the air-flow rate, and the circulating air temperature deviated considerably from those used in an accepted proprietary system.[7] The performance of the panel system is representative only of the warm-air ceiling panel system as installed; it does not necessarily represent that which might be obtained with other transfer media or other locations of the panel. For both systems a single gas-fired forced-air furnace was used.

The following summary of the results obtained in Research Residence No. 2 is applicable to the conditions under which the investigation was conducted.

(1) As far as room-air temperatures and average room-surface temperatures were concerned, the performances of the panel and convection systems were remarkably alike. The panel system produced a wider zone of minimum air temperatures near the exposed wall in the living room than did the convection system, even though the range of temperatures from maximum to minimum was about the same. The room-air temperature differentials in the living zone were less for the panel system than for the convection system.

(2) The mean radiant temperatures determined with a thermo-integrator were slightly higher for the panel system and differed only slightly from the room-air temperatures. The same conclusions were reached when comparisons were made of average surface temperatures for the two systems.

(3) Even though a conventional room thermostat was used for both the convection system and the panel system, no difficulties were experienced with either system in the automatic temperature control. No temperature over-runs or thermal lags were experienced with the panel system under normal operation with a constant thermostat setting.

(4) When the thermostat setting was reduced 10 F for an 8-hr night-time period, the panel system required about twice as much time (5 hr vs. $2\frac{1}{2}$ hr) for recovery to the original room-air temperatures as did the convection system.

(5) The fuel consumption for the panel system was higher than for the convection system. Adequate insulation of the ducts in the attic as well as of the top side and exposed edges of the panel space was shown to be essential.

(6) Although temperature conditions in the first-story rooms were satisfactory with the unheated basement, improvements were noted when the basement was heated, since smaller room-air temperature differentials and warmer floor-surface temperatures were obtained.

(7) When heat was supplied to the basement the fuel consumption was about 10 percent higher than when no heat was introduced into the basement. The corresponding increase in basement-air temperatures was about 7 F.

(8) The results obtained with continuous operation of the blower showed lower fuel consumption but higher electrical consumption than with intermittent operation.

(9) The use of a minimum differential setting for the room thermostat produced remarkably uniform room-air temperatures. For the low air-flow rates used, the minimum differential setting of the room thermostat was more effective for close control of air temperatures than was the fan-switch setting.

(10) For the purpose of selecting the maximum fuel input rate it was concluded that the total heat loss from the structure, including the basement, should be considered regardless of whether or not the basement was heated.

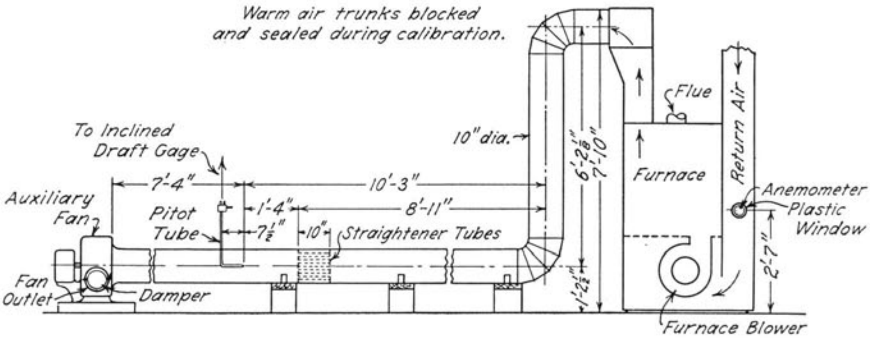
(11) Both direct heat regains (lights, cooking, appliances, sun, and occupants) and indirect heat regains (furnace casing, furnace bonnet, basement ducts, wall stacks, flue pipe, and chimney) affect the actual performance of any heating plant. They are suggested as a possible explanation of the similarity in performance of the panel and convection systems, and also of the deviations between design and performance values for each system. Performance values may be appreciably less than design values, thus providing a margin of safety for extreme weather conditions.

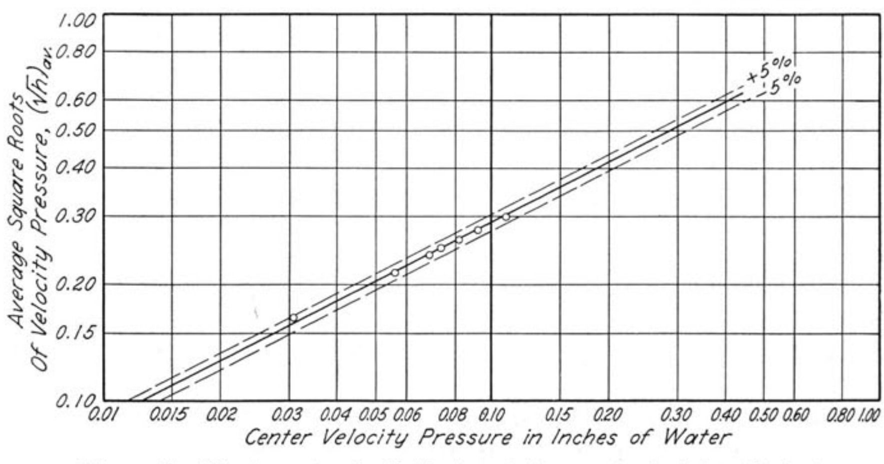
(12) The extended-plenum system provided a satisfactory method of air distribution to the branch ducts.

APPENDIX A
CALIBRATION OF ANEMOMETER IN RETURN-AIR DUCT

40. General Statement

The air-flow rate through the furnace casing and duct system was determined by an anemometer located in the return-air duct of the furnace and calibrated in position. The arrangement of the apparatus used in the calibration is shown in Fig. 28. Prior to the calibration of the anemometer, it was found advisable to calibrate a standard pitot tube in place in the 10-in.-diam duct. For this purpose a curve was





Above: Fig. 28. Apparatus for Calibration of Anemometer in Return-Air Duct
Below: Fig. 29. Calibration Curve for Pitot Tube in 10-in. Duct

established showing the relationship between the velocity pressure at the center of the duct and the average square roots of the velocity pressures over the cross-sectional area of the duct (Fig. 29). The method and procedure for establishing the curve are outlined[14] in Appendix A of Engineering Experiment Station Bulletin 342. After the calibration curve was established, it was only necessary to make observations with the pitot tube located at the center of the measuring station. The average of the square roots of the velocity pressures could be obtained from the calibration curve. The air-flow rate could then be readily determined.

Following these preliminary tests the relationship between the anemometer readings and the air-flow rates was obtained. For this purpose a 4-in. vane anemometer having a dial at right angles to the vanes was placed in the return-air duct of the furnace (Fig. 28). The warm-air branches were blocked and sealed at the furnace so that all of the air passing the anemometer would also pass the pitot-tube measuring station. A viewing panel was located in the side of the return-air duct to permit observations of the anemometer without disturbing the flow of air. A stop watch that could be read to within 0.2 sec was used to determine the time for 3 revolutions of the anemometer.

41. Procedure

Since the anemometer in the furnace was to be used for measuring air-flow rates for both systems, four series of calibration tests were necessary. The test conditions were as follows.

(1) For the convection system, in which both the first story and the basement were to be heated, the return-air duct for the panel system was blocked and sealed.

(2) For the panel system, in which the first-story was to be heated by a ceiling panel and the basement by convection, the first-story return-air ducts were blocked and sealed.

(3) For the 1948-49 season, conditions (1) and (2) were repeated except that branches leading from the basement return-air intakes were also blocked and sealed.

The procedure for calibrating the anemometer in the return-air duct was as follows.

Step 1: The inclined draft gage was leveled and the zero reading adjusted.

Step 2: The auxiliary blower and furnace blower were started and allowed to operate about 15 min before any observations were made. This permitted the motors to operate steadily and the circulated air to attain a constant temperature.

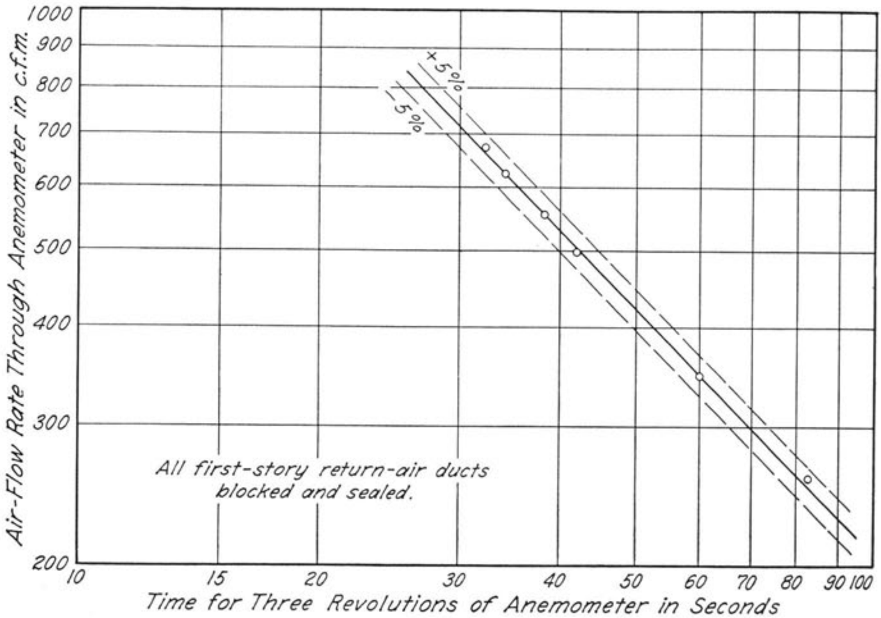


Fig. 30. Calibration Curve for Anemometer in Return-Air Duct (Panel System)

Step 3: With varying settings of the cross damper at the outlet of the auxiliary blower, the center velocity pressure reading of the pitot tube was recorded as well as the time required for 3 anemometer revolutions.

Step 4: For a given velocity pressure at the center of the measuring station, the average of the square roots of the velocity pressures was obtained from the calibration curve shown in Fig. 29. The air-flow rate at the pitot tube station and anemometer was calculated, assuming no air leakage between the two stations.

For each setting of the cross damper this procedure was repeated.

Step 5: The data were plotted on log-log scale paper and a straight line was drawn through the points (Fig. 30). The maximum deviation from the average curve of any point representing an individual traverse did not exceed 3 percent. Similar calibration curves were prepared for the remaining three test conditions. From these curves it was possible by determining the time required for 3 revolutions of the anemometer to read directly the cfm flowing through the system.

APPENDIX B
AIR TEMPERATURE GRADIENTS WITHIN PANEL SPACE

In previous installations of ceiling panel systems (Section 5b), a panel depth of $3\frac{1}{4}$ in. has usually been specified. For the Residence installation, in which open-web steel joists 8 in. deep were used, a shallower panel would have involved the placement of special supports for the top cover of the panel air space. Since the 8-in. depth was simpler to use and no convincing proof existed that it would prove unsatisfactory, the decision was made to use the full 8-in. joist depth for the panel air space and to determine whether this depth would be satisfactory. The use of the 8-in. panel might conceivably cause stratification of air in the panel because of the lower air velocity. With such stratification, higher average air temperatures would be required in order to obtain the necessary heat transfer through the ceiling. For the study of temperature gradients in the panel, thermocouples were installed 2 in. apart on vertical supports located at three points in the south

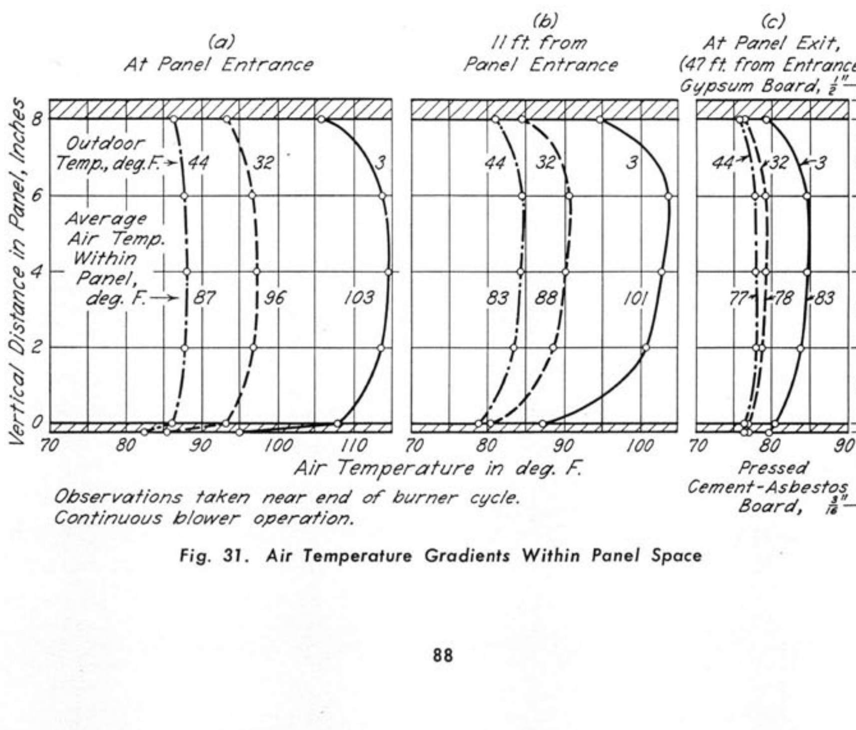


Fig. 31. Air Temperature Gradients Within Panel Space

bedroom panel space. The three points selected were at the panel entrance, at a point 11 ft from the entrance, and at the panel exit 47 ft from the entrance. The south bedroom panel was chosen as a typical panel for these observations since two outside edge exposures existed.

Figure 31 shows the temperature gradients obtained for outdoor temperatures of 44 F, 32 F, and 3 F. The curves indicate that no stratification existed within the panel at any of the three locations. It was concluded that the panel depth of 8 in. was satisfactory.

The temperature drop per foot of air travel within the panel space increased as the outdoor temperature decreased. Figure 31 also shows the average values of air temperature within the panel for each weather condition. The average temperature drop per foot of air travel for the first 11 ft within the panel space was 0.36 F, 0.73 F, and 1.1 F for outdoor temperatures of 44 F, 32 F, and 3 F respectively. Considering the entire length of air travel within the south bedroom panel space, the average temperature drop per foot of air travel was 0.21 F, 0.38 F, and 0.64 F for outdoor temperatures of 44 F, 32 F, and 3 F respectively. These values are in close agreement with the values for the other panel spaces in the Residence.

Similar studies were made for the temperature gradient with the unheated basement. Even though higher panel-air temperatures existed, the gradients were essentially the same.

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